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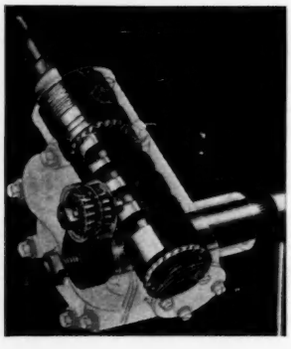

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Cylinder Wear, Where and Why

By S. W. Sparrow and T. A. Scherger
The Studebaker Corp.

THIS paper presents numerous charts showing the wear of cylinder and piston rings. Attention is called to the fact that maximum wear usually is found in line with the top ring at the upper limit of its travel. This is attributed to the presence of dirt and to the high temperatures which make lubrication difficult in this region. Reasons are given for believing that the piston rings are more often the agent producing wear, than is the piston.

Special experiments are described which illustrate the effect of insufficient oil, of abrasives, of rough surfaces and of high local temperatures. It is concluded that inadequate lubrication, but not necessarily insufficient lubricant, is the primary cause of wear.

THE archæologist digs and guesses and even while he proclaims the discovery of the tooth pick of Tutankhamen he is haunted by the thought that this may later prove to be the sword of Cyrus. With somewhat the same feeling this discussion was prepared for it has been a matter of "digging up" records of wear and attempting their interpretation.

Fig. 1 will serve as a starting point. Here, and in general throughout the paper, results are plotted as wear per 10,000 miles and, as a common basis for comparison, one mile is taken as equivalent to 2500 ft. of piston travel. In this instance the piston rings rather than the piston appear to have made the major contribution to wear. For example, maximum wear is found in line with the upper limit of travel of the top compression ring and below the lower limit of ring travel the wear is small. Furthermore considerable wear is found in line with the piston pin although the piston has an initial clearance of about 0.010 in. in this direction. A heavy deposit of burned oil evidences the lack of contact during the test. It should be mentioned, however, that some wear may occur as a result of the contact between this enamel-like oil deposit and the cylinder wall.

Wear of the piston rings is shown in Fig. 2. The width of the wear rectangles indicates the width of the contact surfaces. Toward the end of the test, wear had reduced the tension of the rings to such an extent as to permit considerable blow-by. This further accelerated the rate of wear and was responsible for the breakage of three rings. The top ring in each cylinder wore more than the second ring and less than the oil control ring. Relatively small contact surfaces and high unit pressures are responsible for the rapid wear of oil rings. One reason why the chalk wears faster

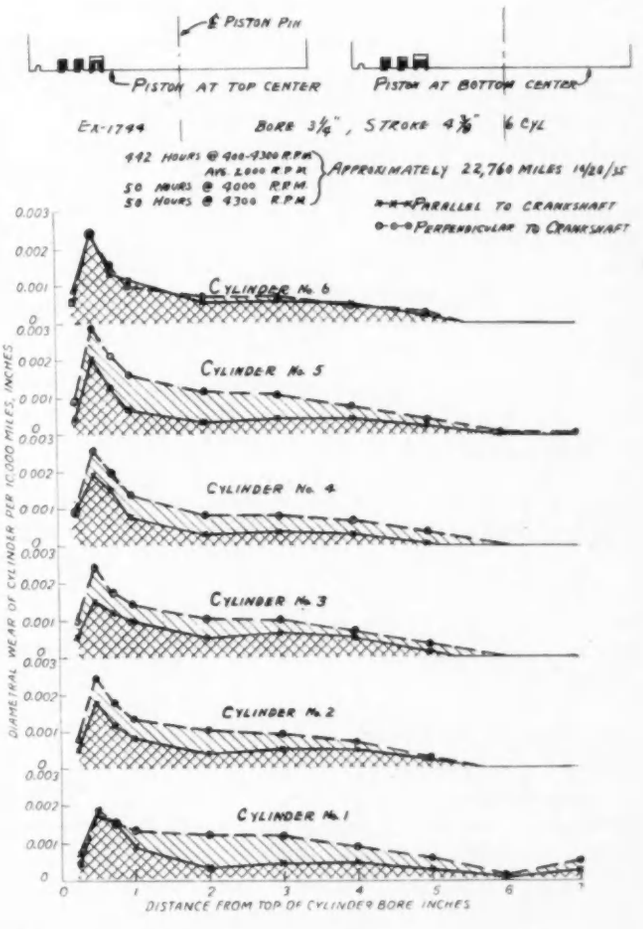
℄ Piston Pin
Piston at top center
Piston at bottom center
Ex-1744
Bore 3 1/4", Stroke 4 3/8" 6 Cyl
442 Hours @ 400-4300 R.P.M. Avg. 2000 R.P.M.
50 Hours @ 4000 R.P.M.
50 Hours @ 4300 R.P.M.
Approximately 22,760 Miles 14/20/55
Parallel to Crankshaft
Perpendicular to Crankshaft
Cylinder No. 6
Cylinder No. 5
Cylinder No. 4
Cylinder No. 3
Cylinder No. 2
Cylinder No. 1
Diametral Wear of Cylinder per 10,000 Miles, Inches
Distance from top of cylinder bore inches

Fig. 1—Note That Maximum Wear Occurs in Line with Top Ring

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 15, 1936.]

Fig. 2—Ring Flutter Due to Wear is Responsible for Breakage of Three Rings

than the blackboard is that a little piece of chalk travels over a lot of blackboard. One should not forget that a little piston ring travels over a lot of cylinder bore.

Piston measurements are tabulated below. In general, figures for piston wear are likely to be less significant than those for piston rings or cylinders. This arises from the fact that the piston distorts somewhat during the initial period of operation and its rate of wear decreases as the area of contact increases. Hence, it is unsafe to predict piston wear on the basis of measurements made after relatively short tests.

Diametral Wear per 10,000 miles (measured perpendicular to the center line of the crankshaft)

Piston No.	1	2	3	4	5	6
Top of Skirt	0.0010	0.0007	0.0009	0.0008	0.0013	0.0012
Center of Skirt ...	0.0006	0.0002	0.0004	0.0002	0.0008	0.0008
Bottom of Skirt ...	0.0004	0.0000	0.0004	0.0002	0.0005	0.0008

Fig. 3 is based upon 250 hr. of full load operation at 4300 r.p.m. and the wear is fairly uniform over the area traversed by the piston rings. Measurements had been made also after 150 hr. Fig. 3-A and Fig. 3-B provide a comparison between the rate of wear during the last 100 hr. and the rate over the full 250 hr. The comparison shows differences but these are not sufficiently consistent to justify drawing general conclusions.

The corresponding wear of the piston rings during the first 150 hr. is found in Fig. 4 and the eye is at once drawn to the abnormal wear of the rings in cylinders 2 and 5. In the hope of finding an explanation for this condition new rings were installed in the first three cylinders. Referring to Fig. 4-A it will be seen that neither the new rings in No. 2 nor the old rings in No. 5 showed excessive wear during the following 100 hr. The abnormal wear of the rings in 2 and 5 is attributed to the main bearings which were used during the last 50 hr. of the 150-hr. period. In these, the grooves governing the oil supply to the crankpins were less than half the normal length. Past experience with engines of this type has shown that crankpins 2 and 5 suffer most from too low oil pressure or too short a period of registry. Obviously a decrease in the amount of oil reaching the crankpins means a decrease in the amount thrown on the cylinder walls. It seems probable, therefore, that *less* oil was responsible for *more* wear of the rings in No. 2 and No. 5.

Despite the statement that "People eat too much," starvation is not an impossibility and while minimum wear and minimum oil consumption are apt to go hand in hand it is well to realize that thrift in the matter of oil may be carried too far. Fig. 5 tells a story which is pertinent at this point. The photograph was taken after one hour of full load operation of 3000 r.p.m. Prior to the test the cylinder walls were given a "sand blast" finish and new piston rings were installed and pinned to prevent turning. The contact surface of the oil ring was filed away over a width of about one inch in line with the center of the thrust face of the piston. Hence, the oil ring was ineffective over this one-inch width. Compression rings were normal.

The wearing away of the frosted surface gives a clearly

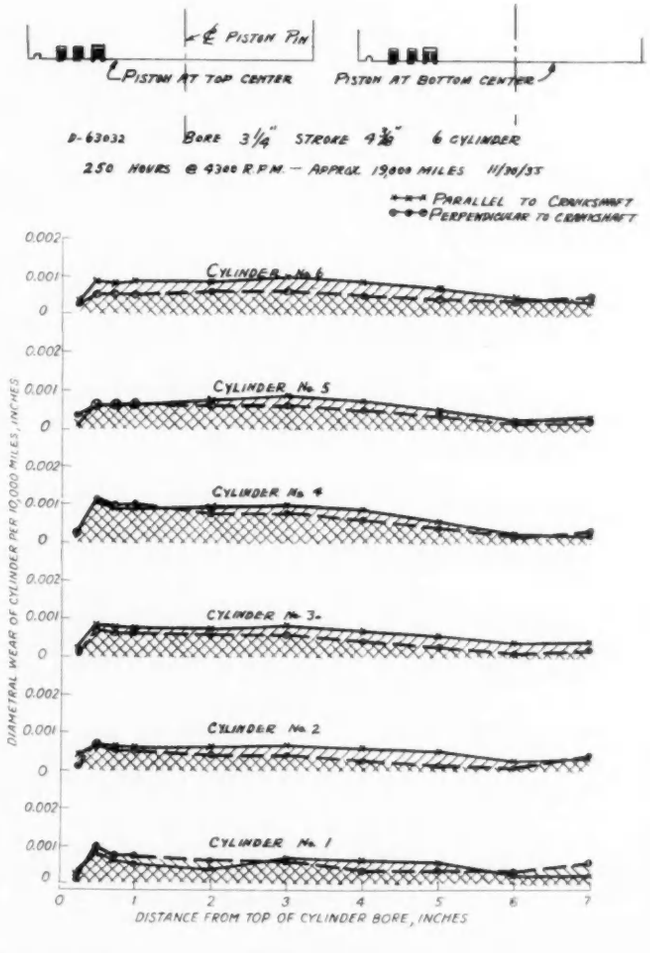
℄ PISTON PIN
PISTON AT TOP CENTER
PISTON AT BOTTOM CENTER
D-63032 BORE 3¼" STROKE 4⅜" 6 CYLINDER
250 HOURS @ 4300 R.P.M. — APPROX. 19,000 MILES 11/30/35
PARALLEL TO CRANKSHAFT
PERPENDICULAR TO CRANKSHAFT
CYLINDER No. 6
CYLINDER No. 5
CYLINDER No. 4
CYLINDER No. 3
CYLINDER No. 2
CYLINDER No. 1
0.002
0.001
0
DIAMETRAL WEAR OF CYLINDER PER 10,000 MILES, INCHES
0 1 2 3 4 5 6 7
DISTANCE FROM TOP OF CYLINDER BORE, INCHES

Fig. 3—Wear Distribution Was Fairly Uniform After High Speed Test.

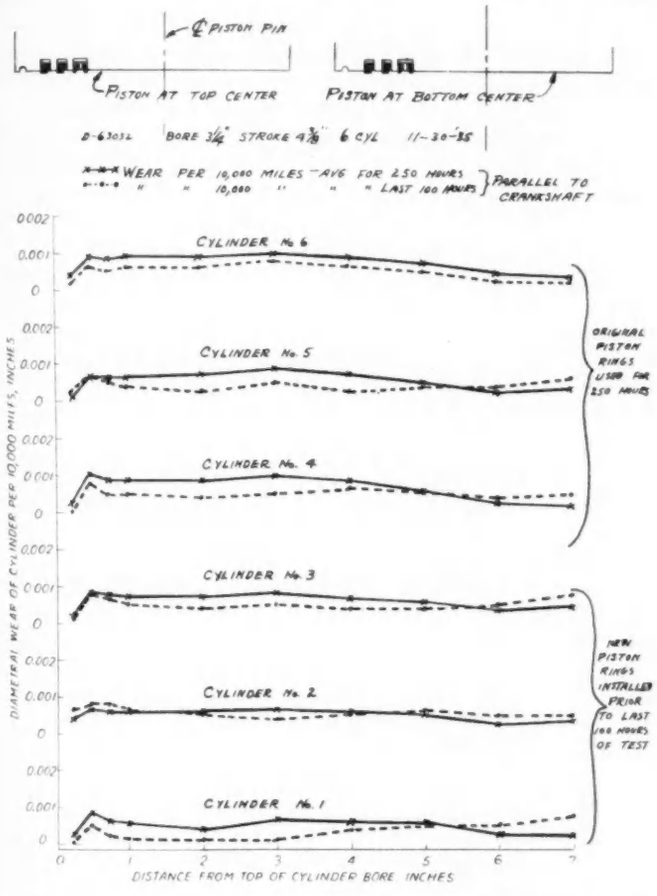
℄ PISTON PIN
PISTON AT TOP CENTER
PISTON AT BOTTOM CENTER
D-63032 BORE 3¼" STROKE 4⅜" 6 CYL 11-30-35
WEAR PER 10,000 MILES AVG FOR 250 HOURS
" " " " LAST 100 HOURS
PARALLEL TO CRANKSHAFT
CYLINDER No. 6
CYLINDER No. 5
CYLINDER No. 4
CYLINDER No. 3
CYLINDER No. 2
CYLINDER No. 1
ORIGINAL PISTON RINGS USED FOR 250 HOURS
NEW PISTON RINGS INSTALLED PRIOR TO LAST 100 HOURS OF TEST
DIAMETRAL WEAR OF CYLINDER PER 10,000 MILES, INCHES
DISTANCE FROM TOP OF CYLINDER BORE, INCHES

Fig. 3A—Comparative Rates of Wear During 250-Hr. Test.

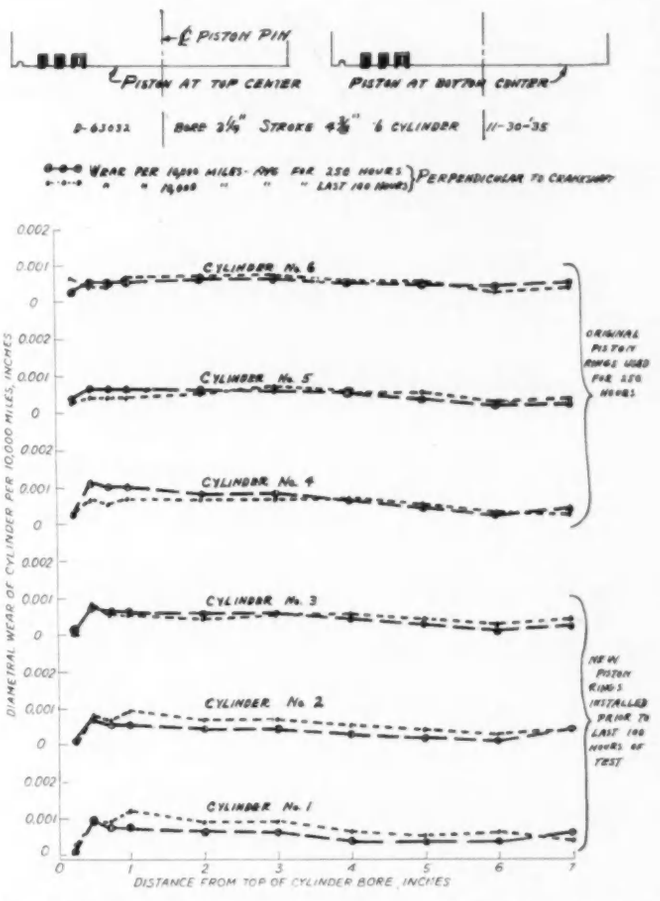
℄ PISTON PIN
PISTON AT TOP CENTER
PISTON AT BOTTOM CENTER
D-63032 BORE 2¼" STROKE 4⅜" 6 CYLINDER 11-30-35
WEAR PER 10,000 MILES AVG FOR 250 HOURS
" " " " LAST 100 HOURS
PERPENDICULAR TO CRANKSHAFT
CYLINDER No. 6
CYLINDER No. 5
CYLINDER No. 4
CYLINDER No. 3
CYLINDER No. 2
CYLINDER No. 1
ORIGINAL PISTON RINGS USED FOR 250 HOURS
NEW PISTON RINGS INSTALLED PRIOR TO LAST 100 HOURS OF TEST
DIAMETRAL WEAR OF CYLINDER PER 10,000 MILES, INCHES
DISTANCE FROM TOP OF CYLINDER BORE, INCHES

Fig. 3B—New Piston Rings May Be Responsible for a Slight Increase in Rate of Wear

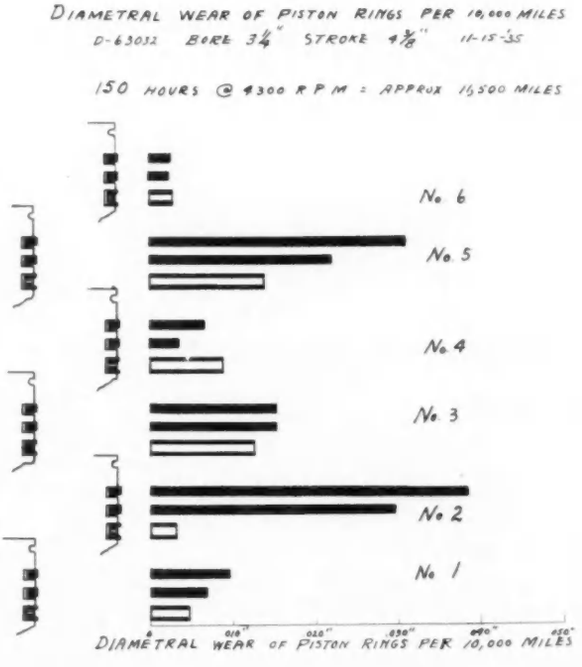
DIAMETRAL WEAR OF PISTON RINGS PER 10,000 MILES
D-63032 BORE 3¼" STROKE 4⅜" 11-15-35
150 HOURS @ 4300 R.P.M. = APPROX 16,500 MILES
No. 6
No. 5
No. 4
No. 3
No. 2
No. 1
DIAMETRAL WEAR OF PISTON RINGS PER 10,000 MILES

Fig. 4—Excessive Wear of Rings in Nos. 2 and 5 Is Attributed To Insufficient Oil

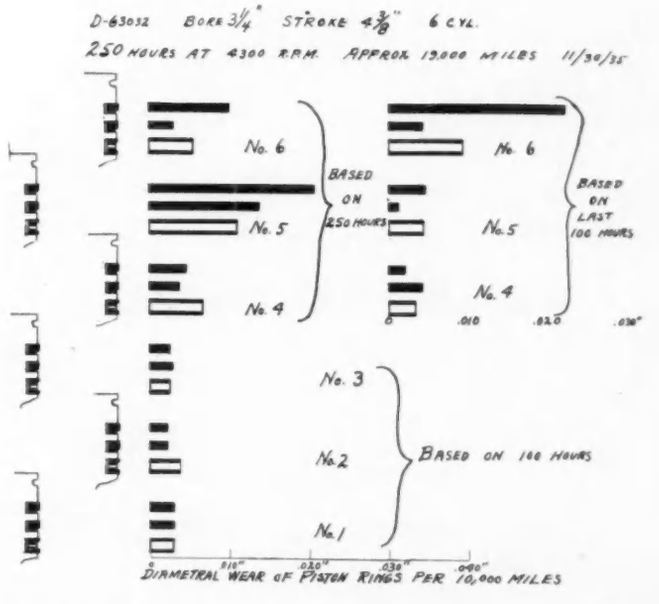
D-63032 BORE 3¼" STROKE 4⅜" 6 CYL.
250 HOURS AT 4300 R.P.M. APPROX 19,000 MILES 11/30/35
No. 6
No. 5
No. 4
No. 3
No. 2
No. 1
BASED ON 250 HOURS
BASED ON LAST 100 HOURS
BASED ON 100 HOURS
DIAMETRAL WEAR OF PISTON RINGS PER 10,000 MILES

Fig. 4A—Note that Rings in Nos. 2 and 5 Do Not Show Excessive Wear During Last 100 Hr.

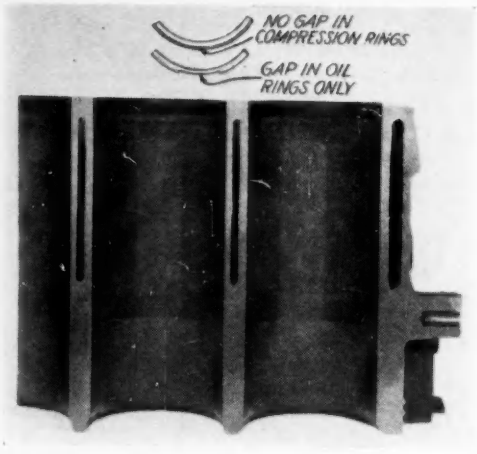

Fig. 5—Absence of Wear in Longitudinal Band Is Due To Gap Filed in Oil Ring

defined record of the regions of maximum wear. The frosted surface has not been worn away in the area beyond the travel of the piston rings nor has it been worn away in line with the filed gap in the oil control ring, even though this surface has been traversed by two compression rings which were

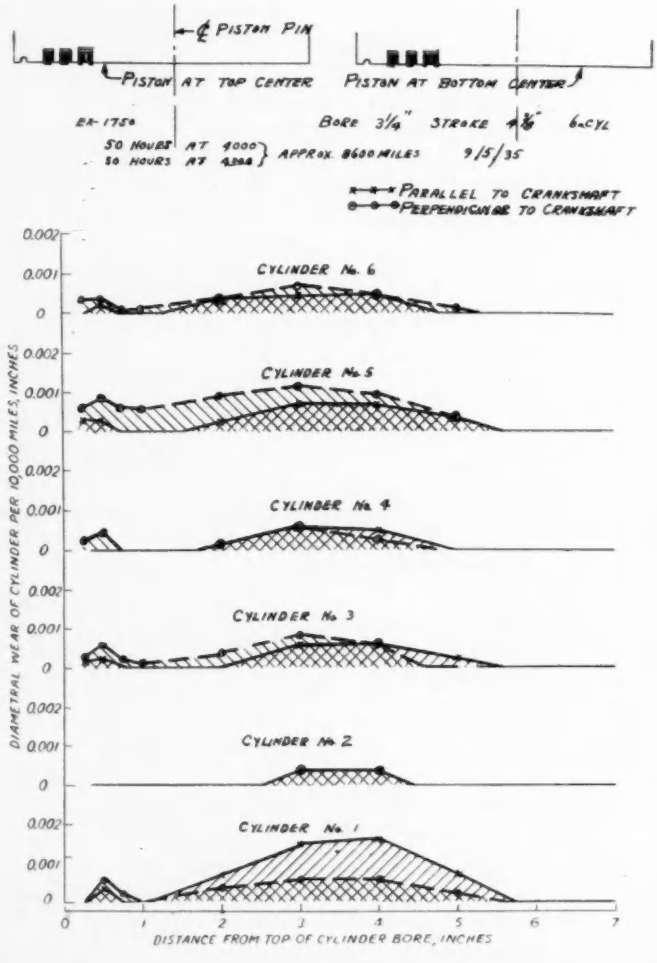

Fig. 6—Wear Characteristics Are Not Normal As Maximum Wear Is Not Usually Found in Center of Bore

normal in every respect. It should be remembered also that the piston exerts its maximum thrust in this unworn area.

At first thought the experiment might seem to point to the metal-to-metal contact between the oil ring and the cylinder wall as the sole cause of wear. Such a theory is shattered when we observe that the worn surface extends to the upper limit of travel of the top compression ring which is well beyond the path of the oil ring. Hence, we are forced to conclude that wear was caused by the compression rings as well as by the oil control ring. The item of chief interest, however, is that the compression rings did not wear through the frosted surface over the one-inch width where the oil control ring was ineffective. Here, thanks to the gap in the

Fig. 7—In This Test There Was Little Difference Between the Wear of the First and Second Rings

oil ring, there was a thick film of oil and the compression rings retained sufficient of this oil on their surfaces to prevent metal-to-metal contact throughout the entire stroke. It must be remembered that the surface of the bore was purposely roughened and that for this reason the rate of wear was abnormally high. Nevertheless, the experiment emphasizes the fact that the lubrication of the compression rings is dependent upon the oil left on the walls by the oil control ring and that compression rings which receive insufficient oil because of a too effective oil control ring may be a major cause of wear.

Returning again to Fig. 4-A, we have been unable to find a satisfactory explanation for the rapid wear of the top ring in No. 6 during the last 100 hr. of the test. Further proof of our inability to "fathom the unfathomable and unscrew the inscrutable" comes with Fig. 6. Here we find the wear to be a maximum near the center of the bore. This is unusual and as yet unexplained. Piston wear averaged less than 0.001 in. per 10,000 miles and was a maximum for No. 2 which happens to be the cylinder in which the wear was a minimum. As for the rings, Fig. 7 shows nothing abnormal.

Fig. 8 and Fig. 9 require no special comment other than to mention that the test consisted of 300 hr. of full load

operation at 4500 r.p.m. and to point out that the wear of the compression rings was extremely small.

Results of a road test of about 60,000 miles appear in Fig. 10. Measurements were not taken over the entire bore but wear was fairly uniform over the portion traversed by the skirt of the piston and averaged less than 0.0002 in. per 10,000 miles. As usual the wear was a maximum near the top of the cylinder. For the last 30,000 miles new rings were installed and the "top" ring groove was left vacant. Hence there were two "top" ring positions during the test and this explains why there is so little difference between the wear 0.2 in. and 0.4 in. from the top of the block.

Faulty water distribution coupled with a rough bore should be blamed for the excessive wear of the rings in the three rear cylinders of Fig. 11. This condition had been improved prior to the test shown in Fig. 12 but the cooling of No. 6 was still inadequate. In both tests scuffing of the cylinder walls testified to the lack of cooling. Where cylinder wall temperatures are abnormally high the viscosity of the oil is decreased and the piston rings break through the film. This is illustrated in Fig. 13. Two sets of rings were pinned and operated for one hour in the same cylinder. Average water outlet temperatures were 170 deg. fahr. and 205 deg. fahr. No doubt the temperatures in some parts of the jacket were somewhat higher. In any event, wear was negligible with the lower water temperature and excessive with the higher temperature. Fig. 14 shows a piston from which the tin

EX-1733 3 1/16 BORE 4 1/4" STROKE 8 CYLINDER
300 HOURS @ 4500 RPM APPROX 23000 MILES 2/25/35
No. 8
No. 7
No. 6
No. 5
No. 4
No. 3
No. 2
No. 1
DIAMETRAL WEAR .010" OF PISTON RINGS .020" .030" .040" PER 10,000 MILES

Fig. 9—Note Low Rate of Wear of Compression Rings

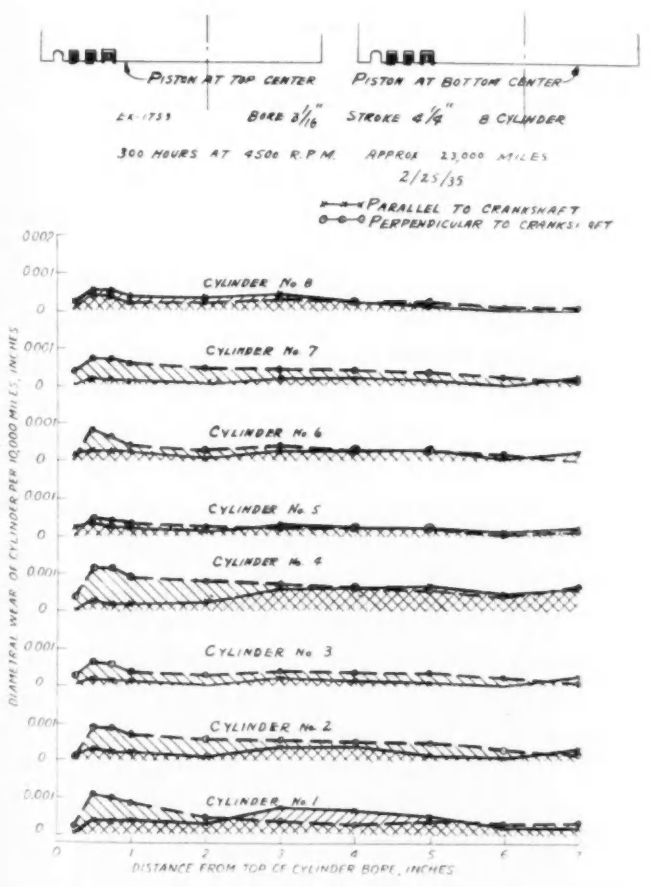
PISTON AT TOP CENTER
PISTON AT BOTTOM CENTER
EX-1733 BORE 3 1/16" STROKE 4 1/4" 8 CYLINDER
300 HOURS AT 4500 R.P.M. APPROX 23,000 MILES
2/25/35
PARALLEL TO CRANKSHAFT
PERPENDICULAR TO CRANKSHAFT
CYLINDER No. 8
CYLINDER No. 7
CYLINDER No. 6
CYLINDER No. 5
CYLINDER No. 4
CYLINDER No. 3
CYLINDER No. 2
CYLINDER No. 1
DIAMETRAL WEAR OF CYLINDER PER 10,000 MILES, INCHES
DISTANCE FROM TOP OF CYLINDER BORE, INCHES

Fig. 8—Cylinder Wear Resulting from Continuous High Speed Operation

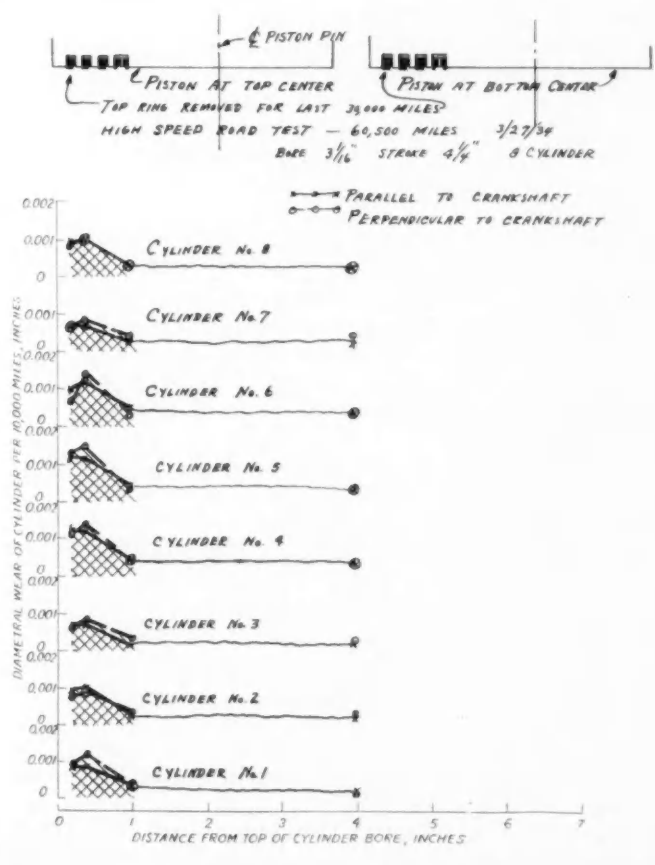
PISTON PIN
PISTON AT TOP CENTER
PISTON AT BOTTOM CENTER
TOP RING REMOVED FOR LAST 30,000 MILES
HIGH SPEED ROAD TEST — 60,500 MILES 3/27/34
BORE 3 1/16" STROKE 4 1/4" 8 CYLINDER
PARALLEL TO CRANKSHAFT
PERPENDICULAR TO CRANKSHAFT
CYLINDER No. 8
CYLINDER No. 7
CYLINDER No. 6
CYLINDER No. 5
CYLINDER No. 4
CYLINDER No. 3
CYLINDER No. 2
CYLINDER No. 1
DIAMETRAL WEAR OF CYLINDER PER 10,000 MILES, INCHES
DISTANCE FROM TOP OF CYLINDER BORE, INCHES

Fig. 10—Cylinder Wear from High Speed Road Test

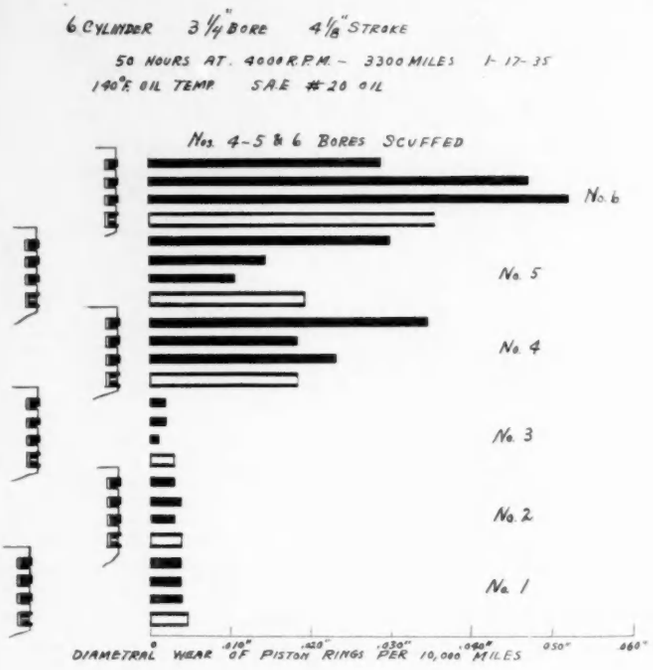
6 CYLINDER 3 1/4" BORE 4 1/8" STROKE
50 HOURS AT 4000 R.P.M. - 3300 MILES 1-17-35
140°F OIL TEMP. S.A.E. #20 OIL
Nos. 4-5 & 6 BORES SCUFFED
No. 6
No. 5
No. 4
No. 3
No. 2
No. 1
DIAMETRAL WEAR OF PISTON RINGS PER 10,000 MILES
.010" .020" .030" .040" .050" .060"

Fig. 11—Incorrect Water Distribution Caused Excessive Wear of Piston Rings in Nos. 4, 5 and 6

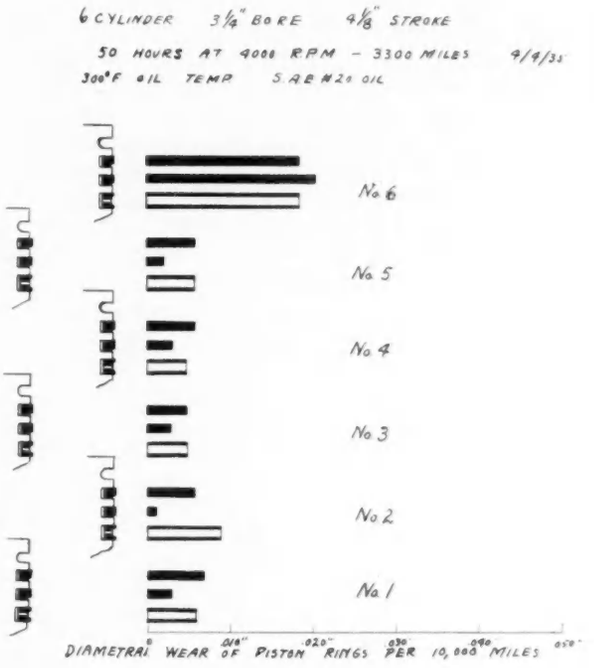
6 CYLINDER 3 1/4" BORE 4 1/8" STROKE
50 HOURS AT 4000 R.P.M. - 3300 MILES 9/9/35
300°F OIL TEMP. S.A.E. M20 OIL
No. 6
No. 5
No. 4
No. 3
No. 2
No. 1
DIAMETRAL WEAR OF PISTON RINGS PER 10,000 MILES
.010" .020" .030" .040" .050"

Fig. 12—Excessive Wear of Rings in No. 6 Bore Due To Incorrect Water Distribution

plating has been worn away by contact with an inadequately cooled portion of the cylinder wall.

The next three figures are based on road tests. The fact that the pistons were of cast iron with the oil control ring at the bottom of the skirt has no particular bearing on the results. Fig. 15 contributes a more or less normal record with maximum wear as usual in line with the top center position of the top compression ring. In Fig. 16, however, "things are not what they seem" as will be evident when it is observed that the wear is plotted in terms of *1000* instead of *10,000* miles. Although the engine had been returned with the complaint that the iron was soft, the actual cause of the excessive wear was insufficient protection against dirt. Sand has the modern urge to go places and do things and in Fig. 17 it was encouraged to make the engine its destination. The encouragement took the form of a pipe which replaced the air cleaner and had its entrance beneath the running board of the car. Its ability to do things is attested by the fact that the wear is plotted in terms of *100* miles. A better picture of the relative wear is offered by Fig. 18, where results from all three tests are plotted to the same scale.

The dirt test is of interest not because the wear is excessive but because the general characteristics of wear are the same as in normal operation. Note for example that in Fig. 18 the rate of wear per 1000 miles decreases from 0.030 in. at the upper end of the cylinder to about 0.003 in. at the lower end. Note also, in Fig. 19, the high rate of wear of the oil ring and top compression ring in comparison with the rates of the intermediate rings. The inference is that despite such watch dogs as air cleaners and oil filters a considerable amount of abrasive either enters the combustion chamber or is formed therein. As to the latter possibility, the top land of the piston in Fig. 20 bears testimony to the effectiveness of hard carbon as an abrasive. Wherever the top of the piston is free from carbon the top land is free from scratches.

The low rate of wear of the second compression ring in comparison with the top ring in the presence of abrasive can be attributed to the protection furnished by the top ring and by the piston lands above. These serve as a first line of defense and tend to pulverize the abrasive until the maximum dimension of many of the particles is less than the thickness of the oil film. Thus in Fig. 21, a coarse abrasive has wrought havoc with the lands above the top ring but has left unscathed the carbon deposit upon the lands below. A companion piston was operated at the same time without compression rings and Fig. 22 shows that all of the lands

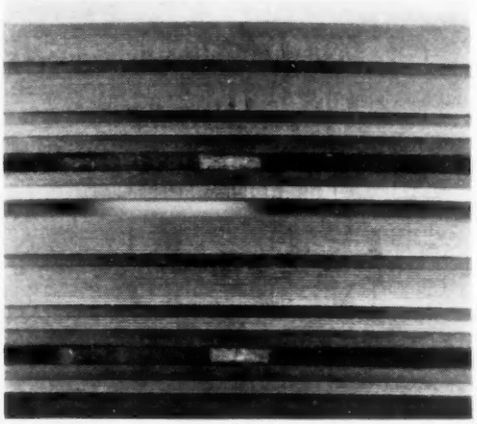

Fig. 13—Ring Scuffing Caused by Hot Spot in Cylinder Wall

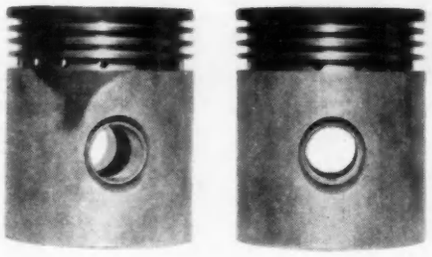

Fig. 14—Tin Plating Worn Because of Local Hot Spot

were scratched. In both instances the top land suffered more than the lands below.

Abrasive which enters through the intake valve may eventually depart through the exhaust valve, having worn out its welcome as well as a bit of the cylinder wall. Abrasive in the form of a rough spot on the surface of the bore is more in the nature of a permanent paying guest with payment being made in the form of sly digs whenever the rough spot comes in contact with a piston ring. The two sets of rings of Fig. 23 saw service in the same engine and in cylinders which supposedly had the same finish. Presumably the scratches on one set of rings came from a rather small rough spot, with the rotation of the rings causing the scratches to be widely distributed. Had the rings been pinned, the rough spot would have dug fewer and deeper ditches. It is difficult to say which condition is the worse but it is easy to realize that both are undesirable.

The importance of maintaining a film of oil on the cylinder walls during the starting period is well understood and it will suffice to mention a couple of experiments merely as a reminder. The preparation for the first experiment consisted in installing new piston rings in two cylinders of an engine and then washing the piston, rings, and walls of one of the cylinders with gasoline. The experiment itself consisted of cranking the engine for 5 min. at 130 r.p.m. The product is found in Fig. 24 and Fig. 25—a scuffed piston, a badly worn oil ring and badly scratched compression rings, all as the result of a slow speed, 5-min. journey in a dry cylinder. In the next experiment four cylinders were washed with gasoline and then five starts were made at 10-min. intervals. Air temperature was about 70 deg. fahr. and the engine was permitted to make but a few revolutions after each start. The reward for this effort was the rust which is seen in the dry bores in Fig. 26. The rust on one set of rings is shown in Fig. 27. Naturally, there was no rust in the cylinders which were properly lubricated.

No attempt will be made to discuss the interesting work

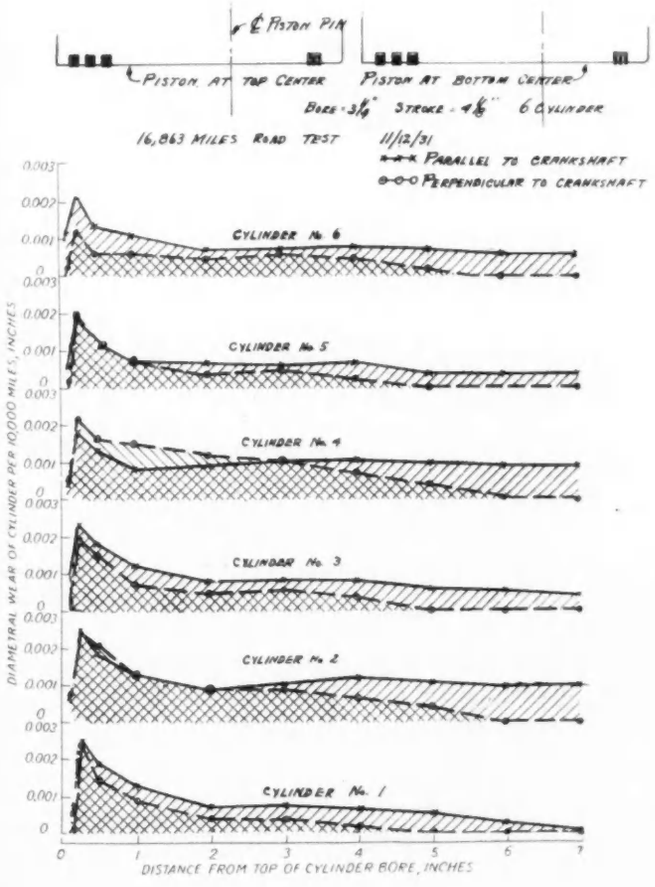
℄ Piston Pin
℄ Piston at top center
Piston at bottom center
Bore 3 1/4" Stroke 4 1/8" 6 Cylinder
16,863 miles Road Test 11/12/31
Parallel to crankshaft
Perpendicular to crankshaft
Cylinder No. 6
Cylinder No. 5
Cylinder No. 4
Cylinder No. 3
Cylinder No. 2
Cylinder No. 1
Diametral wear of cylinder per 10,000 miles, inches
Distance from top of cylinder bore, inches

Fig. 15—Road Test with Normal Protection Against Dirt

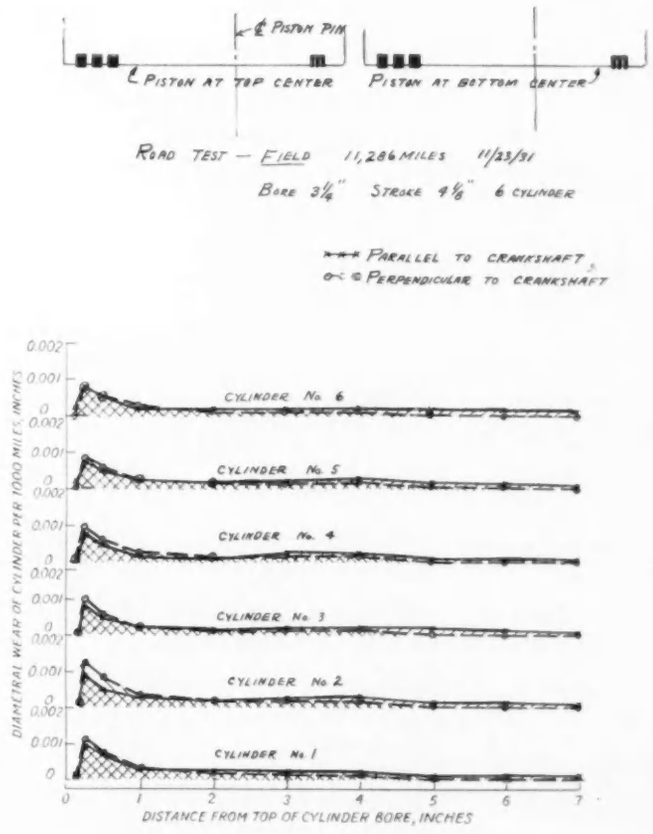
℄ Piston Pin
℄ Piston at top center
Piston at bottom center
Road Test — Field 11,286 miles 11/23/31
Bore 3 1/4" Stroke 4 1/8" 6 Cylinder
Parallel to crankshaft
Perpendicular to crankshaft
Cylinder No. 6
Cylinder No. 5
Cylinder No. 4
Cylinder No. 3
Cylinder No. 2
Cylinder No. 1
Diametral wear of cylinder per 1000 miles, inches
Distance from top of cylinder bore, inches

Fig. 16—Dirt Caused Excessive Wear

on corrosion recently reported by the Institution of Automobile Engineers, nor to comment upon the many other factors whose effect on wear, though important, is not clearly indicated in the records upon which this report is based. From the data at hand, however, we may answer the question implied in the title of this paper by stating that wear is found *where* the lubrication is inadequate and to the "why" we may say that it occurs *because* the lubrication is inadequate. Occasionally inadequate lubrication is synonymous with insufficient lubricant but more often it is due to abrasives, to blow-by which destroys the lubricant, to diluent or to local high temperature areas which reduce the viscosity of the oil, or to rough surfaces or regions of high unit pressure which break through the film.

And to the "where and why?" of the title might well be added "what of it?" For after all the real goal is freedom from the harmful effects of wear, and durability by itself is not always a satisfactory criterion of merit. Increasing the tension of a compression ring may increase wear at certain engine speeds and yet the increased effectiveness in the control of blow-by may prevent excessive wear and even more serious consequences under other conditions of operation. The best oil control ring is not the one which wears least but the one which gives satisfactory oil control for the longest time. Nor

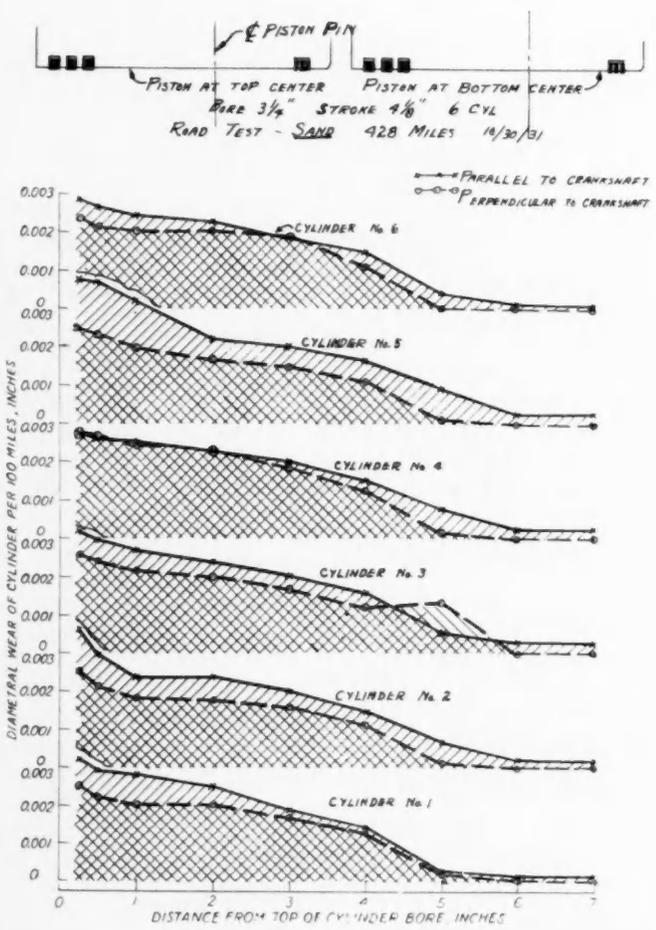
℄ PISTON PIN
PISTON AT TOP CENTER
PISTON AT BOTTOM CENTER
BORE 3¼" STROKE 4⅛" 6 CYL
ROAD TEST - SAND 428 MILES 10/30/31
PARALLEL TO CRANKSHAFT
PERPENDICULAR TO CRANKSHAFT
CYLINDER No. 6
CYLINDER No. 5
CYLINDER No. 4
CYLINDER No. 3
CYLINDER No. 2
CYLINDER No. 1
DIAMETRAL WEAR OF CYLINDER PER 100 MILES, INCHES
DISTANCE FROM TOP OF CYLINDER BORE, INCHES

Fig. 17—Dirt Fed To Engine

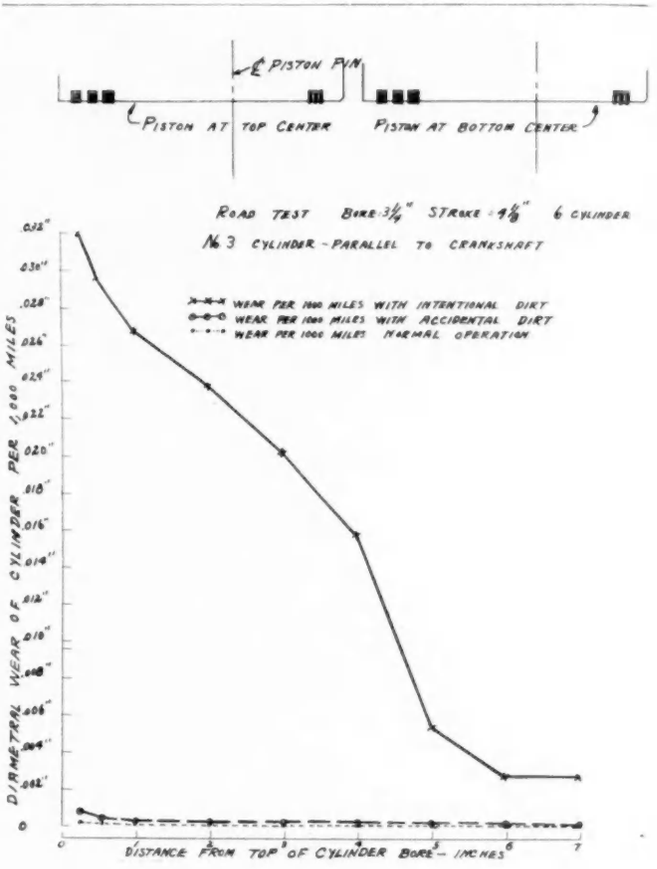
℄ PISTON PIN
PISTON AT TOP CENTER
PISTON AT BOTTOM CENTER
ROAD TEST BORE 3¼" STROKE 4⅛" 6 CYLINDER
No. 3 CYLINDER - PARALLEL TO CRANKSHAFT
WEAR PER 1000 MILES WITH INTENTIONAL DIRT
WEAR PER 1000 MILES WITH ACCIDENTAL DIRT
WEAR PER 1000 MILES NORMAL OPERATION
DIAMETRAL WEAR OF CYLINDER PER 1,000 MILES
DISTANCE FROM TOP OF CYLINDER BORE - INCHES

Fig. 18—Influence of Dirt on Wear

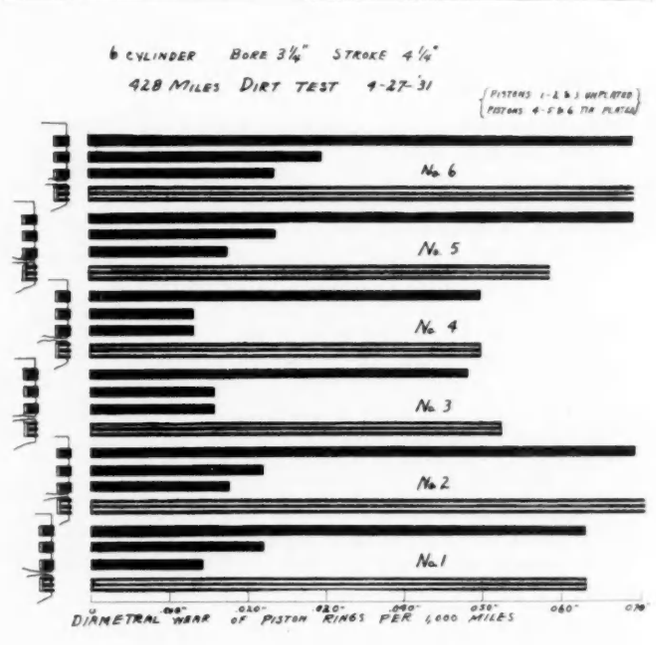

Fig. 19—Abrasive In Combustion Causes Excessive Ring Wear

is the piston which serves silently for the most miles necessarily the piston which wears least. Other things being equal, minimum wear is desirable but it behooves us to make certain that other things are equal.

The archæologist digs and guesses with an ever increasing realization of how much still remains hidden. This discussion was begun with feelings akin to those of an archæologist. With similar feelings it is concluded.

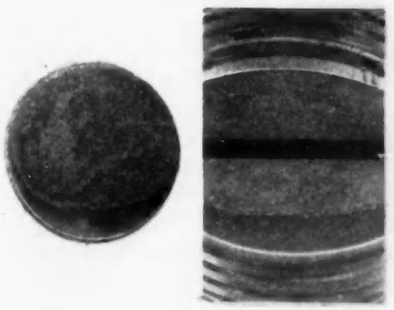

Fig. 20 — Hard Carbon Causes Erosion of Top Land

Fig. 21 — The Top Piston Ring Protects the Lower Lands and Rings

Fig. 22 — All Lands Eroded in Test Without Compression Rings

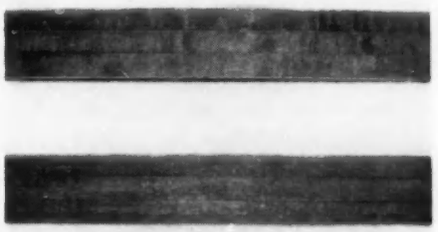

Fig. 23—Piston Ring Scuffing Caused by Rough Spot on Cylinder Bore

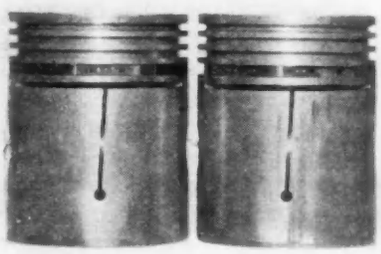

Fig. 24—Piston Scuffed From Lack of Oil

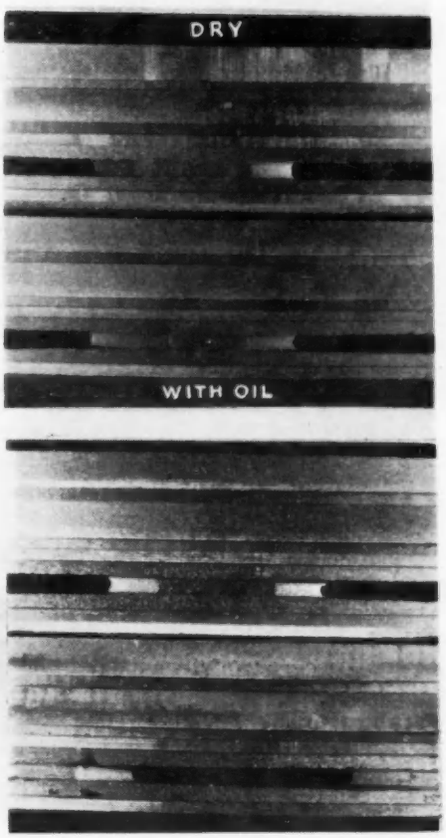

Fig. 25 — Rings Scuffed From Lack of Oil

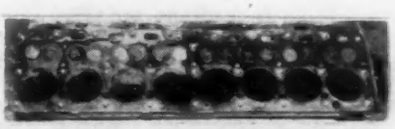

Fig. 26 — Rust Caused by Lack of Protective Oil Film

Fig. 27 — Rusting of Rings as a Result of Insufficient Oil

Air Springs—Tomorrow's Ride

By Roy W. Brown

In Charge of Research Engineering, The Firestone Tire & Rubber Co.

THE results of an extensive research in motor-car suspension fundamentals are presented with a description of a new-type spring member using air as the load-carrying means.

Methods of actually realizing extremely low spring rates with attendant comfort to the passenger are shown. Shock absorption and anti-body-roll devices are incorporated in, and function as integral parts of, the new suspension.

A series of new tools are presented for the chassis engineer's consideration in designing better riding comfort into the new car. These include in a simple, practical light-weight unit, variation of spring rate with wheel position, approximate aperiodic body damping, body roll reduced to better than that secured with high-rate springs, wide range of load capacity with practically constant body frequency, and ability to adjust wheel position to the optimum point.

Typical applications are illustrated together with numerous curves outlining the stated characteristics.

Laboratory tests that have indicated phenomenal flexing life have been confirmed by extensive road tests. Measurements, confirmed by observation under service conditions, substantiate the realization of riding-comfort improvements indicated as possible by the low air spring rates. Curves presented show how this has been accomplished without loss of adequate shock absorption and cornering stability.

THE problem of protecting the vehicle passenger from the uncomfortable and fatiguing effects of road roughness is as old as the wheeled vehicle itself. Although much progress has been made in reducing the number and the height of road-surface irregularities, the best road surface modern science has attained is of such nature as to cause discomfort and fatigue, especially during long trips at high speeds. These conditions have made necessary the use of some device in the vehicle between the road and the passenger which will minimize the effects of road roughness.

Motor Car Suspension Requirements

Maintained high speeds for several hours with today's motor cars are now quite common. Safe operation at such speeds under the traffic conditions encountered dictates the extensive and complex mechanical conditions that must be fulfilled by any practical suspension, in addition to provision for such characteristics as will result in the greatest comfort and least fatigue for the passenger.

[This paper was presented at the Annual Meeting of the Society, Detroit, January 14, 1936.]

[1] See *Engineering*, September 18, 1925, p. 367; extracts from "The Main Vibration of an Autocar," by Prof. James J. Guest, Institution of Automobile Engineers.

[2] See S.A.E. TRANSACTIONS, September, 1935, pp. 313-321; "'Sky-Hooks' for Automobiles," by W. S. James, H. E. Churchill, and F. E. Ullery.

[3] See S.A.E. TRANSACTIONS, March, 1934, pp. 73-81; "Independent Wheel Suspension—Its Whys and Wherefores," by Maurice Olley, also, July, 1934, pp. 248-256; "Dynamics of the Modern Automobile," by George L. McCain.

[4] *See Journal De La Société Des Ingénieurs De L'Automobile*, June, 1933, pp. 2242-2253; "Les Voitures a Roues Independantes," by Maurice A. Julien.

An excellent review of the underlying theory of motor car suspensions appeared in 1925.[1] Results of an extensive study of methods of practical application of this theory were published in 1935.[2] The effect of weight distribution and other physical suspension characteristics was quite thoroughly studied[3] at the time that independent wheel suspensions were introduced to the American market. Numerous types of independent wheel suspensions have been used in Europe[4] since the early days of automobiles. Several of these treatises appear to offer a possible solution to the mechanical requirements. Literature is replete with many other articles, a resumé of which is startling in the inconsistencies advocated. About the only conclusion that can be drawn from them is to the effect that the suspension of a motor car is tremendously complex both in theory and application.

Three major factors may be segregated which concern the passenger, as their combined effect largely determines the extent of his physical and mental discomfort and fatigue resulting from road shocks. These are:

(1) *Spring Rate.*—This factor determines the extent of shock transmission to the passenger and of the direct wheel and chassis motions caused by road irregularities.

(2) *Shock Absorption.*—Provision is necessary to diminish succeeding motion of the chassis caused by stored energy in the suspension and moving parts after the wheel has passed over the road irregularity.

(3) *Body Roll.*—Horizontal forces developed in rounding corners acting at the center of the sprung weight cause an

outward angular rotation of the body on the suspension system which becomes greater with lower-rate springs.

The first of these factors can be minimized by using soft springs in the suspension, that is, springs in which a small additional load results in appreciable deflection. The second condition is met by shock-absorber damping devices, the effect of which must be varied for the softer springs. In a number of instances the third condition of body roll in cornering has been minimized by fitting a member between wheels, either front or rear, in such a manner that elastic torsional deformation of the member occurs when movement of either wheel with reference to the body is different than the opposite wheel, which condition occurs with angular motion of the body.

Unfortunately, the inertia and highly viscous drag of the hydraulic shock absorbers extensively used at present result in restriction of wheel movement, the reaction to which is transmitted to the passenger as a series of definite and sharp vibrations. Likewise, the torsional member fitted between the wheels very appreciably minimizes the advantages of a softer suspension when the motion of the two wheels to which the member is connected is other than equal both in direction and extent. Measurements show such an unequal condition to predominate in actual road service.

The compromises involved in fitting adequate shock-absorbing and torsional-stabilizing devices have, in effect, prevented the passenger from fully realizing the comfort possible with soft springs.

Air Spring Development

With such a background a research was undertaken some time ago that has been successful in evolving a simple, practical means of carrying the total load on a low-rate air-inflated bellows with self-contained shock-absorbing and anti-body-rolling properties. The new suspension is shown diagrammatically in Fig. 1 and as installed on the front wheels of a motor car in Fig. 2.

The development of a serviceable bellows proved a problem of no small magnitude in the application of rubber. The success realized is attested by the bellows' ability to withstand eight million, 2-in. deformations while under a 25 per cent overload. Road tests have been equally favorable in indicating a flexing life appreciably beyond that heretofore demanded for suspensions.

Direct measurement of the more important factors has correlated conflicting opinions as to the relative significance of the various complex factors involved. In this manner efficiency has been introduced to research, followed by advantages in cost, time, and results secured.

Spring Rate

The deflection which a spring undergoes when loaded, of course, can be conveniently measured in the laboratory. However, factors such as inter-leaf friction, clamping pressure, and shackles in the case of leaf springs; and lever-arm ratio and friction in the linkage joints in the case of independent springing, make necessary actual measurement if usable results are to be secured of spring rates on the motor car, as pointed out by Maurice Olley.[3] The results of a number of such tests on representative present-day motor cars are shown in Fig. 3.

Since the forces to which the passenger is subjected, due to road irregularities, are the result of the combination of spring rate and friction, reduction of friction becomes very important when it is considered that friction approximates spring rate in the lower-rate suspensions now in service. The matter of friction has been the subject of extensive study which has been quite well summarized by Georges Broulhiet.[5]

The boulevard ride requires extremely low-rate springs. Wheel movement under this condition is small compared to that occurring at higher speeds, or at abusive speeds, over rough roads. The ideal spring rate to meet both of these

[5] See S.A.E. TRANSACTIONS, October, 1933, pp. 325-347; "Independent Wheel Suspension," by Georges Broulhiet.

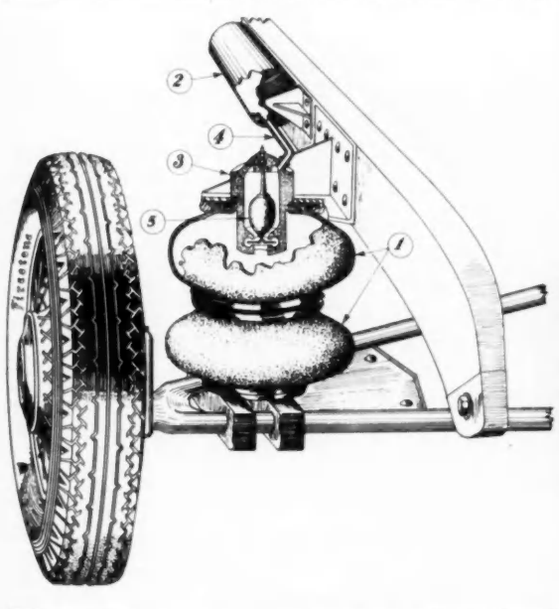

Fig. 1—Air Spring Diagram—Complete Suspension

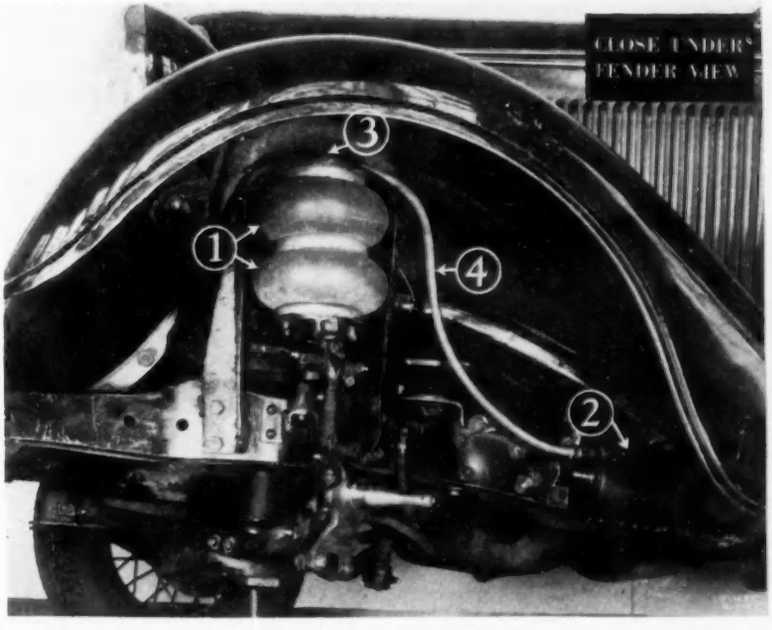

Fig. 2—Air Spring Application

The Firestone air spring consists of a rubberized fabric bellows (1), inflated with air to carry the desired load, communicating with an air reservoir (2) through the valve (3) and tubing (4). Movement of the pendulum (5) makes available the higher rate of the bellows for resisting "body roll" and "nosing down." The valve also provides any desired shock absorption action.

conditions is one in which the rate increases both on compression and recoil from the normal load position. The elastic deformation of the metal used in conventional springs gives equal rate at all loads. Numerous mechanical devices[6] have been devised to compensate for this characteristic of metal springs. Apparently they have failed to come into use because the many loaded joints necessary result in a large friction effect and a high cost due to their comparatively complicated mechanisms.

The air spring has been designed so as to approximate the ideal rate condition necessary for the boulevard ride and also to provide the increase in rate desirable to prevent bottoming and excessive recoil when driving at high speeds or over rough roads. The outstanding difference in these two types of rate curves is evident from inspection of Fig. 4. Design of the bellows can be varied to provide higher or lower rates as desired.

The size of reservoir used with each air spring determines the static rate characteristics. Thus the suspension engineer can select from a curve similar to that shown in Fig. 5 the particular reservoir size which best meets the characteristics of the car upon which it is to be applied.

The air spring has an inherent advantage in that its dynamic rate characteristics can be varied over and above those previously described by restricting the passage between the bellows, indicated by a (1) in Fig. 1, and the reservoir, indicated by a (2). This adjustment can also be made by means of inertia, velocity, or remote manually operated valves if desired, thus providing another device with which to work to adapt the particular job to the best possible riding-comfort condition.

Damping—Shock Absorption

It is regrettable that no complete solution has so far appeared to the problem of damping the wheel and body movements resulting from stored energy after passing over road irregularities. Until a means can be found to attach, in effect, damping devices between the wheel and the road, it is probable that inadequate damping of the wheel movement will continue. In the better grade of electrical instruments perfect damping is realized through use of inherent parts of the movement itself. This method suggests at least a more complete solution to the shock absorption problem on motor cars.

Damping on the compression stroke (wheel moving upward) while the body is moving upward tends to aggravate the movement. Later in the body-movement cycle when the body is moving downward, compression damping tends to decrease the downward body motion. The first effect is undesirable, but the second is desirable. Damping on the recoil stroke (wheel moving downward) when the body is moving upward tends to minimize the body motion, while damping tends to increase the body motion when the body movement on this stroke is downward. In this case, the first effect is desirable, and the second is undesirable. When employed on front-wheel springing, damping during the recoil stroke has a further disadvantage in that, when a wheel leaves the road, it interferes somewhat with steering by increasing the length of time the wheel remains out of contact with the road surface. The damping problem is further complicated by the inability thus far to apply adequate damping devices independently to the comparatively high-frequency tire oscillations.

Adequate damping devices, as previously pointed out, must be free from friction, inertia, and highly viscous drag and, preferably, operating characteristics should not change with temperature variations. The work cycle of an adequate damping device approaches that of the spring itself, indicating another advantage for a spring construction with inherent damping properties.

In the air spring this construction is accomplished by inserting a valve, shown by a (3) in Figs. 1 and 2, between the

[6] U. S. Patents 1,800,485; 1,833,238; and 1,897,538.

SPRING CHARACTERISTICS – 1936 MOTOR CARS

TEST NUMBER	FRONT				REAR			
	WHEEL LOAD LBS (W)	RATE LBS/IN (S)	FREQUENCY CYCLES/MIN (F)	FRICTION LBS.	WHEEL LOAD LBS (W)	RATE LBS/IN (S)	FREQUENCY CYCLES/MIN (F)	FRICTION LBS.
LIST PRICE $500-$700 (4-DOOR SEDAN)								
2	776	182	91	145	735	142	83	95
9	680	180	97	90	753	106	71	113
11	767	124	76	105	821	117	71	105
15**	762	134	79	180	774	125	76	80
$700-$1000								
3	793	110	71	120	813	110	69	100
4	795	169	87	95	780	116	73	110
6*	855	100	64	65	934	110	64	100
10**	721	110	74	156	868	98	63	120
12**	847	82	59	75	854	112	68	85
13	862	147	78	100	801	105	68	125
$1000-$2500								
5*	977	96	59	50	966	118	66	90
7**	884	66	52	45	951	98	60	120
8	1158	240	86	140	1282	153	65	100
MAX	1158	240	97	156	1282	153	83	120
MIN	680	66	52	45	735	98	60	85
AVER	838	134	75	105	872	116	69	103

* FRONT TORQUE STABILIZER NOT INCLUDED
** REAR TORQUE STABILIZER NOT INCLUDED

$F = \frac{88}{\sqrt{\frac{W}{S}}}$

Fig. 3—Average Spring Rates at Normal Wheel Loads

Friction is the weight required to start deflections in an opposite direction from the previous spring movement. This force is transmitted in part to the passenger. Frequency may be considered for intercomparison, as an approximate riding-comfort index, the lower values giving much greater freedom from road shocks.

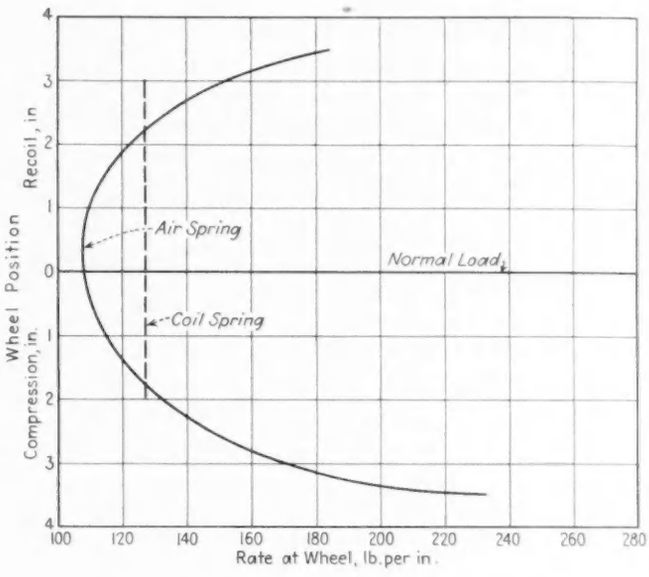

Fig. 4—Air Spring Compared with Coil Spring—Rate Vs. Wheel Position

Comparatively low rate at normal-load wheel position and increasing rate on both compression and recoil is necessary to prevent bottoming and to provide control at high speeds or upon rough roads. Note the extent to which this result is accomplished, as shown, for the suspension illustrated in Fig. 2. Air spring No. 6, serial No. 30. Coil spring (standard equipment) free length, 12¾ in.; O.D., 5⅛ in.; 0.676 in. round; 8.6 free turns. Uncorrected for wheel-control linkage. No bumpers.

bellows and the reservoir. This valve, shown diagrammatically in Fig. 6, consists of a disc (6) arranged to permit free flow of air from the bellows to the reservoir, but to restrict return flow, as desired, through ports (7). In this manner the energy available for recoil damping is represented by the difference between the rates of the bellows with the reservoir size used, and the bellows without reservoir.

Experience has proved that the energy available for damping shown in Fig. 7 is more than sufficient to realize perfect aperiodic damping of the body without measurable interference with steering. Such damping is appreciably more than has been generally used, on account of the interferences which have formerly resulted with the boulevard ride and wheel action at high speeds. In fact, the additional damping secured by this method appears to have improved the rear-seat ride through minimizing pitching by an amount appreciably greater than could have been anticipated from the simultaneous reduction of front-end spring rate.

The physical and mental reactions of the passenger are determined not only by the amount of body movements, but also by their frequency, or the number of times per minute succeeding movements occur. The body of a motor car constitutes a mass suspended on a spring, which system has a natural frequency depending upon the body weight and the amount this weight deflects the springs.

Each year the frequency considered most comfortable has been lowered. Some years ago, when the body frequencies resulting from the suspensions then used were in the neighborhood of 140 cycles per min., a frequency of 126 was thought to be ideal. At present some of the lowest rate suspensions result in body frequencies of from 90 to 110 cycles per min., while 60 to 80 cycles per min. is considered ideal. With true aperiodic damping there is no succeeding cycle. Hence there is no frequency, as the displacement from road irregularity or other cause is limited to one cycle. Apparently this condition accounts for the fact that the larger amount of correct type damping used in the initial experimental installation of the air spring on cars lacking today's weight distribution and stiff frames produced appreciably superior results in the rear-seat ride.

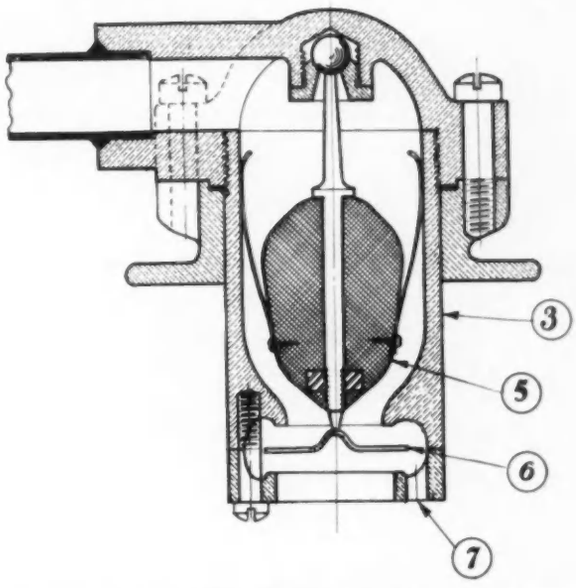

Fig. 6—The Pendulum Valve

Body roll and shock absorption are controlled by this simple valve. The pendulum (5) swings outwardly when rounding turns permitting valve disc (6) to close upwardly, thus producing the higher rate of the bellows available for resisting body roll. Similar action prevents nosing down from brake applications. Recoil control results from valve (6) closing downwardly, thus delaying the return of air from the reservoir by directing the flow through the restricted port (7).

Such damping also eliminates the possibility of exciting resonant vibrations in other parts having equal natural frequencies. This advantage is of particular importance in the event that it is desirable to use front and rear springs under loading conditions which would result in equal natural frequencies if conventional springing were employed.

The smooth damping action available from air flowing through a closed system can be augmented if desired both on compression and recoil, as previously stated, by adjustments employing inertia, velocity, or manually operated valves. Pro-

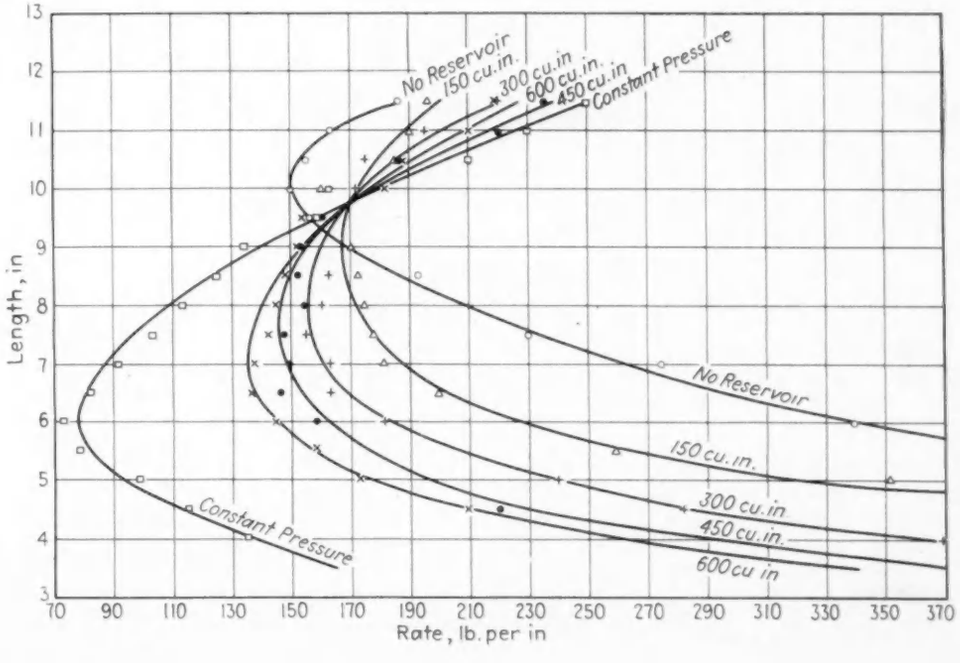

Fig. 5—Air Spring Rate vs. Length

From this family of typical air spring curves can be selected the reservoir size giving the most desirable rate characteristics for the particular suspension under consideration. Lower or higher rates may be secured as desired from different types of air spring constructions. Type No. 6, serial No. 4, load, 800 lb. Inflation pressure 28.8 lb. per sq. in. at 7½-in. length.

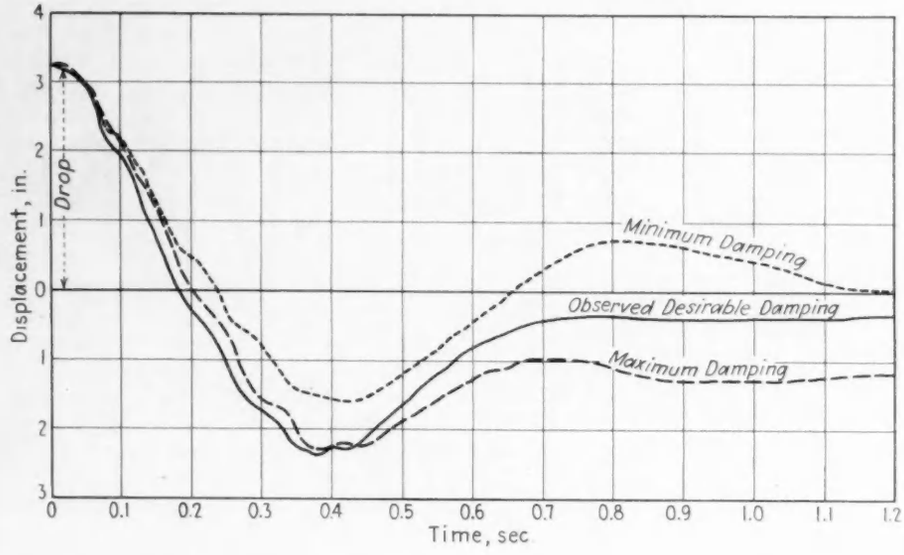

Fig. 7—Drop Test Indicating Amounts of Body Damping Possible with the Air Spring

The car is raised 3 in. and dropped, and the resulting body movement shown is recorded on a chronograph.

portioning of the reservoir and attention to valve design can be employed to damp wheel motion to the maximum extent permitted by steering and body-ride interference. The comparative simplicity of the parts involved permits quick and inexpensive adaptation to provide the most desired results for the individual suspension under consideration.

Body Roll

When a car passes around a curve, centrifugal force acting at its center of gravity tends to displace the body along a line extending radially outward from the center of the curve. Since the tires restrict lateral displacement, rotation of the body takes place dependent in degree upon the characteristics of the suspension. This phenomenon has been the subject of much study and has been briefly summarized by P. M. Heldt.[7] If spring rates are reduced to provide a soft ride, angular rotation of the body in cornering becomes so objectionable as to limit definitely further reduction in spring rate. This condition has been minimized in some conventional suspensions by installing a torsional member between either front or rear wheels, or both. Unfortunately, such a torsional member definitely increases the effective spring rate, except when wheel action is equal both in direction and amount, a condition seldom encountered on the average road.

[7] See *Automotive Industries*, July 21, 1934, pp. 84-86; "Car Sway with Different-Type Front-End Suspensions," by P. M. Heldt.

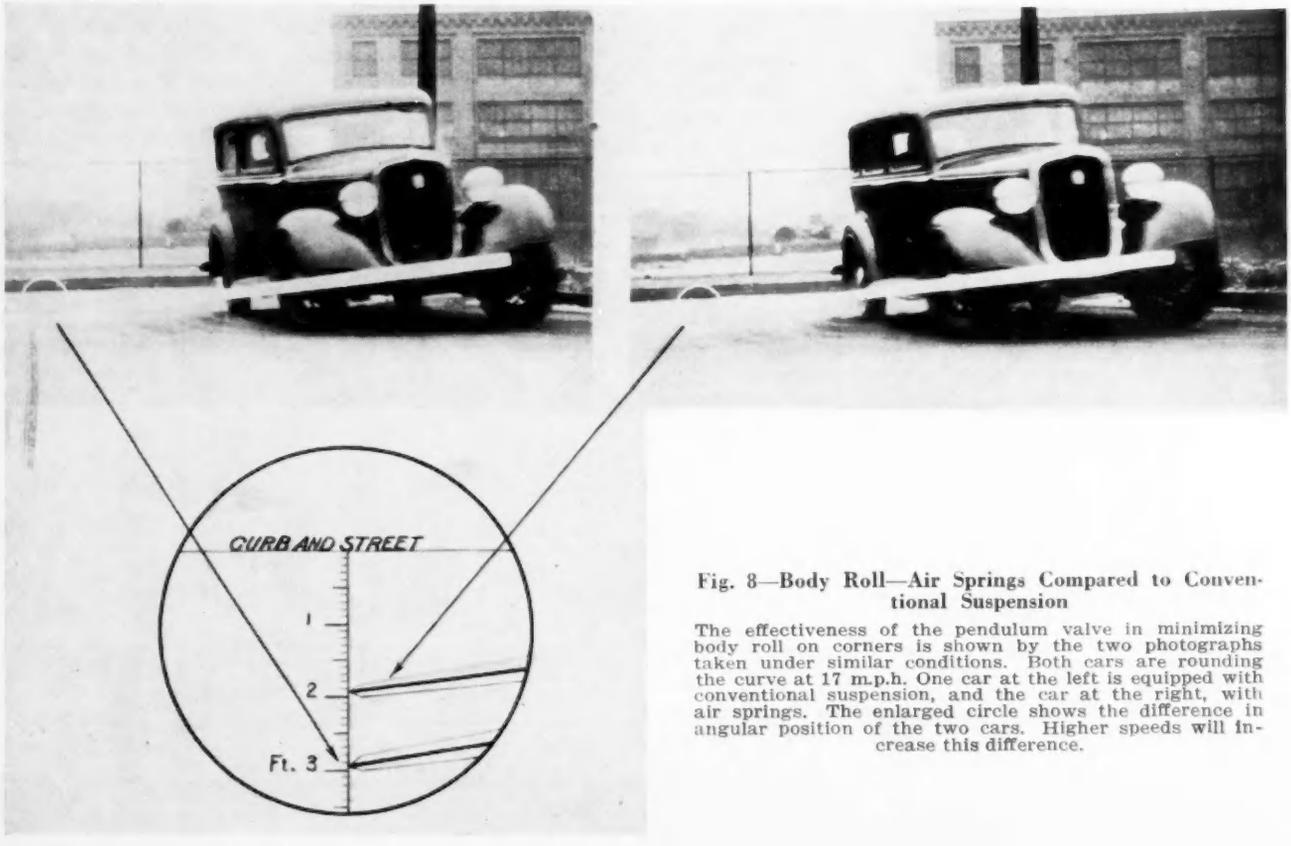

Fig. 8—Body Roll—Air Springs Compared to Conventional Suspension

The effectiveness of the pendulum valve in minimizing body roll on corners is shown by the two photographs taken under similar conditions. Both cars are rounding the curve at 17 m.p.h. One car at the left is equipped with conventional suspension, and the car at the right, with air springs. The enlarged circle shows the difference in angular position of the two cars. Higher speeds will increase this difference.

In the air spring a complete solution to the cornering problem has been found without auxiliary equipment and without increasing the effective rate of the spring when the car is going straight ahead. A vertical pendulum, (5) in Fig. 6, is mounted in the valve (3) so as to swing outwardly when rounding turns. This permits valve disc (6) to close the passage to the reservoir. This action effectively increases the pressure in the bellows because of the additional load resulting from the centrifugal force developed in turning. This pressure prevents the car body from inclining to the outside an undesirable amount when driven on curves at high speeds. The effect is definite and positive, as illustrated in Fig. 8, in reducing body roll to the minimum desirable for optimum driving conditions.

In practice the raised portion of the valve disc (6) is shaped so that the pendulum also disengages when it swings forward, as is the case in quick brake application. This action greatly minimizes the nosing down of the front portion of the car so noticeable in quick brake applications on cars with low-rate conventional front suspensions.

The energy available to accomplish both of these results is represented by the difference in rates of a bellows with reservoir and the bellows alone. Reference to Fig. 5 indicates the rate to be 136 lb. per in. for a typical case with a 7½-in. bellows length, a wheel load of 800 lb., and a 600-cu. in. reservoir, while the rate without the reservoir is 234 lb. per in.

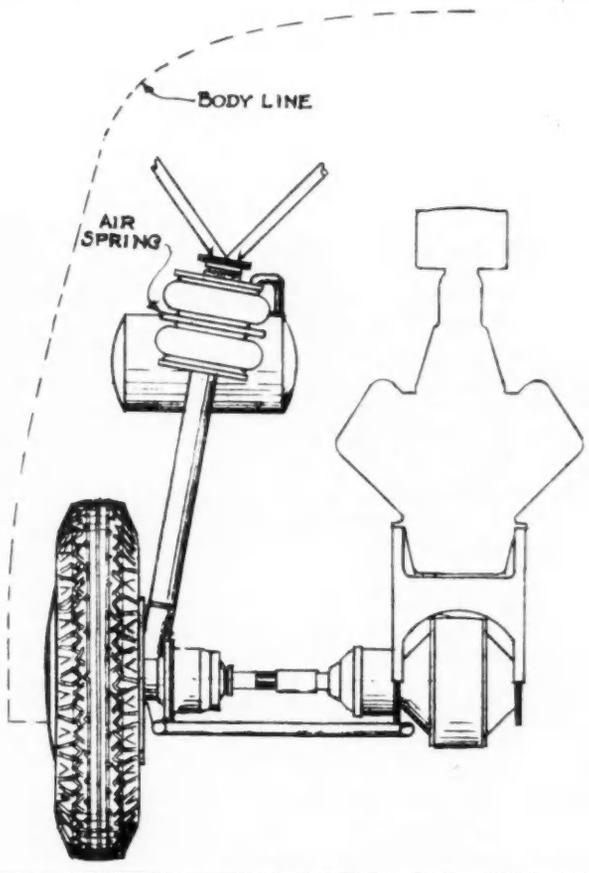

Fig. 9—Air Spring on Stout *Scarab*

Installation of the air spring on this rear-engine car required a low rate in combination with a high load capacity. Low unsprung weight necessitated unusually large shock absorption requirements. These characteristics with the high mounting possible resulted in phenomenal stability under all driving conditions.

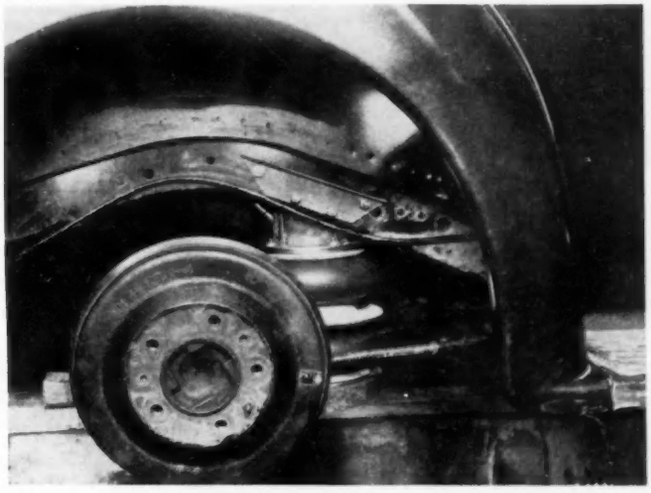

Fig. 10—Under-the-Fender View of a Typical Installation of the Air Spring on a Solid Rear Axle.

Thus a rate increase of 98 lb. per in. is available to resist undesirable rolling and nosing-down reactions.

Air Spring Installation

The exactness in steering geometry demanded by present-day high speeds has caused the development of numerous ingenious mechanisms to locate the wheels positively with reference to the frame structure. Also, much more effective braking has introduced torques at the wheels which are not satisfactorily handled by conventional forms of springs. Abroad, positive wheel control has assumed many forms, mostly independent types. Here, five definite types of independents have appeared. Positive mechanical stabilization of solid axles merits consideration as a possible method of securing the major advantages of the independents with appreciable reduction in production cost.

In Fig. 2 the air spring is illustrated installed on an extension of the upper end of the knuckle forging of a standard front-end parallelogram construction. Mounting the bellows in this position appreciably reduces the load on the wishbone pivots, thereby minimizing wear and lubrication difficulties at these points.

Because air spring bellows are capable of handling large travel with comparatively small overall length and still retain very low rates near the mid point of the travel, the unit can be applied in such locations as to carry the load directly, or with the minimum amount of intervening linkage.

It is logical to expect the car of tomorrow to be equipped with this suspension giving "Tomorrow's Ride". In Fig. 9 is illustrated the rear air spring installation on the Stout *Scarab*. Experience has demonstrated the previously outlined advantageous characteristics of the air spring in this unique application. The outstanding wheel-positioning mechanism used on this car enables installation of the bellows in the engine compartment considerably above the center of gravity. This arrangement results in remarkable stability and shock-absorption properties.

On rear axles of the solid type the bellows may be installed directly between the axle and the frame member, or offset either to the front, as shown in Fig. 10, or to the rear of the axle as desired.

The axle must be mechanically located in a horizontal plane which design, as pointed out previously, is rapidly becoming a necessity for other reasons.

Characteristics of wheel-positioning mechanisms will, of course, be dependent largely upon chassis design. The physical dimensions and load-carrying characteristics of the air spring bellows are such as to provide the chassis designer with a new latitude in applying the load-carrying means to any wheel-positioning device.

A method of laterally locating the axle with torque-tube drives is illustrated in Fig. 11. This construction consists of a single triangular radius rod pivoted at the front on the horizontal axis of the torque-tube ball joint. The rear is pivoted to the torque tube below its centerline. All pivots are rubber bushed. Such construction enables individual vertical wheel movements without accelerating the body laterally, as occurs with conventional solid-axle constructions.

Compression bumpers may be installed within the bellows if desired, although the bellows itself is capable of acting as a recoil bumper. The advantage of eliminating numerous moving joints is particularly noticeable in rear-solid-axle installations of air springs.

Unsprung Weight Reduced

The air spring bellows with means of attachment to the axle will weigh approximately 2 lb. per wheel. This weight may be compared with from 10 to 55 lb. per wheel for steel springs. The advantages of reducing unsprung weight are quite generally realized and need not be further discussed here, except to add that the lighter unsprung weight that formerly was desirable is now becoming a necessity due to present-day driving conditions.

The application of mechanical wheel-positioning devices may add some weight to certain types of construction. The larger portion of such additional weight would, however, contribute to the sprung rather than to the unsprung values. It is probable that the chassis engineer will find methods for adapting the general design to the new suspension that will result in still greater weight reduction.

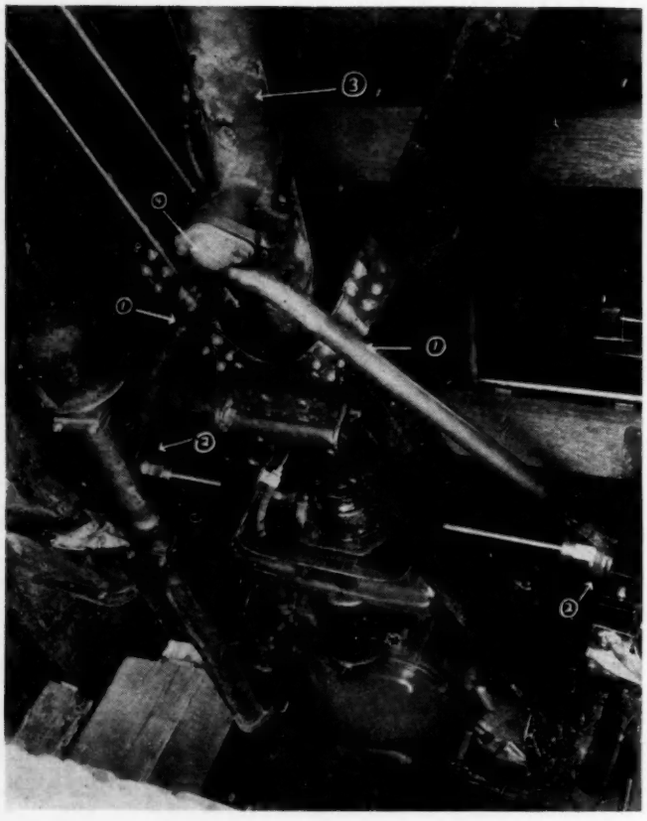

Fig. 11—Under-the-Car View of One Method of Laterally Locating the Rear Axle

The triangular radius rod (1) is rubber bushed (2) at the front end and attached at the rear to the torque tube (3) below its centerline with a spherical rubber bushing (4).

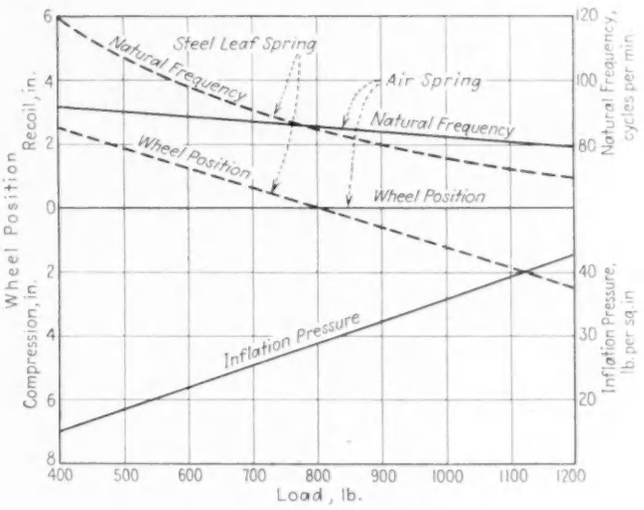

Fig. 12—Typical Air Spring Characteristics

The curves show the ability of the air spring to maintain nearly constant frequency over a wide load range with the wheel maintained continuously in the optimum position for maximum riding comfort under all service conditions. Wheel travel, 3½ in. from normal position. Air spring, Type 6 bellows, vertical length 7½ in., weight 2 lb. Steel leaf spring, horizontal length 56 in., weight 52 lb.

New Tools for the Ride Engineer

In addition to the new properties available in rate, shock absorption, and body roll, the air spring suspension has some other characteristics which can be utilized effectively on designs where other factors impose limitations such as wheel travel, or on entirely new designs where these factors can be controlled to provide the optimum ride condition.

Examination of used as well as new cars with conventional suspensions indicates that the normal static load position of the wheel varies appreciably from the most desirable position. Since riding qualities are definitely limited by wheel travel, this condition means that an appreciable portion of the riding quality possible is lost to the passenger. With the air spring this condition can be corrected by inflating it to the correct wheel position, irrespective of load or other contributing factors.

Within limitations imposed by design considerations of other parts, a suspension should produce the lowest possible natural frequency of body oscillation. Obviously this frequency should remain constant under varying service conditions. From Fig. 12 it is seen that, by varying its inflation, a single air spring bellows design can be utilized for the entire range of wheel loads encountered in passenger cars, without appreciable variation in the natural frequency of the body and without any variation from what is predetermined as the optimum normal-load wheel position. These inherent characteristics provide the chassis engineer with new, simple, and practical material with which to work in providing the passenger with greater comfort and less fatigue. In fact, it appears that the motor car is about to graduate from its mechanical suspension limitations and ride on air as truly as does its transportation relative, the airplane.

The Operator's Woes on Lubrication

By F. L. Faulkner

Automotive Engineer, Armour & Co.

TO the operator, lubricants fall into three general groups: engine-crankcase oil, gear oil, and chassis lubricants. I have elected to discuss them in the order named.

In approaching the subject of automobile engine oils, I do so with a profound respect for the vast amount of research work that has been done in connection therewith, especially that accomplished by the Lubricants Division of the Society. It is not my intention to even attempt to take the time to review here today the voluminous amount of information available on the subject, which in the main is in agreement. It is my only desire, if possible, to bring out for discussion such phases of the problem of engine lubrication that have a direct bearing on the efficient operation of motor-vehicle fleets.

In my last report covering, among other things, starter-battery difficulty, data were given showing the very close tie-in of crankcase lubricant to this problem. Many cases have come to my attention recently where the starter deficiency has been met temporarily by resorting to a crankcase oil having special characteristics. Corrections of this order on a large scale are impractical and costly to the operator.

Reverting for a moment to some of our early operations, we recall that low viscosity oils were then in very general use, and that through the advancing years there has been a definite trend toward higher viscosity oils to provide better piston seal and oil economy. These higher viscosity oils offered no serious operating difficulties, on account of the fact that we were not confronted with what we term the winter problem in those days, as equipment was housed in warm garages and operated on short routes. Operators today, however, are driving their equipment harder and farther than ever before. This condition not only applies to commercial and private fleets but to individual operators as well. Improved highways generally and better facilities on the part of our various highway departments to keep roads open during the winter in northern areas, is responsible for the enormous increase in year around operations.

Influences of Seasons and Locations

In many of our northern operations in years past the motor equipment was laid up during the winter months, and we resorted to other means of transportation. This made the crankcase problem border on the summer side. Today, however, we have a distinct summer and winter problem. Any amount of our equipment along the Gulf and Pacific Coasts operates under summer conditions the year 'round, where temperatures are always above 40 deg. fahr. In the North the summer condition covers about six months from May 1, to Nov. 1, and winter operations are carried on in sub-zero temperatures where the lubricating requirements are extremely exacting.

In general, we should have an oil that is free-flowing at low temperatures. It must be economical from the standpoint of consumption, and yet it must lubricate satisfactorily; it must be free from ring-sticking tendencies and maintain piston seal, and last but not least, it must have a low susceptibility to oxidation.

In our examination of many oils we find that, to meet the requirement of pour test, the matter of viscosity has been seriously affected, which influences the consumption requirement seriously. In order to effect a compromise that will hold consumption to reasonable limits, we are forced to use a viscosity that gives starting difficulties at even moderately low temperatures. In my previous report on starting requirements it was pointed out that the increase in torque required for low temperature cranking was in excess of 40 per cent from 0 deg. fahr. to —15 deg. fahr. Starter-battery combinations as furnished and used with manufacturers recommended S.A.E. crankcase oil are generally inadequate and, when it is realized that their efficiency drops rapidly in service, the operator's troubles are multiplied.

Complications of "Mixed Fleet"

The majority of operators today, particularly those of National scope, have what we term a "mixed fleet," comprised of several makes and many models of the same make of vehicles. Estimating conservatively that the average age of this equipment would approach five years, a great deal of advancement has been made in both engine design and lubrication in this time. The operator, however, is expected to lubricate this conglomeration of equipment in the most efficient manner possible. At the very best this task necessitates many compromises, all of which have a direct bearing on the individual unit's efficiency of operation. When you recognize that the average fleet consists of vehicles ranging from passenger cars to heavy-duty trucks, having all types of lubrication systems for which manufacturers recommend every S.A.E. Viscosity Number from 10 to 60, you will recognize the operator's problem from the service standpoint, keeping in mind that it is impractical to keep available in all garages a wide variety of S.A.E. Viscosity Number oils.

In working out some of these compromises on viscosity we have encountered some very interesting facts. For example, if one make of vehicle that leaves the factory with an S.A.E. 40 oil in summer is drained immediately and put on S.A.E. 10 oil, it will generally operate economically with this grade for quite some time, although occasionally it is necessary to go to S.A.E. 20 oil but rarely heavier. On the other hand if this job that was started off on S.A.E. 40 oil is allowed to see some service before attempting to change to a lower viscosity, the consumption of the lower viscosity oil will, in all cases, be excessive. There is only one conclusion that an operator can draw from this experience, that either the manufacturer has not paid sufficient attention to his original specifications, or he has some other reason for using high viscosity oil that is not apparent. We on the service side are looking

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wholly at economy of operation, which includes both performance and reduced wear-out factors.

I subscribe personally to using the lightest oil you can hold in the crankcase without excessive consumption. Any amount of our heavy-duty equipment is operating on 20-W oil in winter, and S.A.E. 30 oil in summer. Special consideration, however, must be given to specification of these lighter oils, particularly those now refined under new processes. Experiences that I have had up to date with these new lighter oils indicate that there is a greater sludging tendency than with the heavier grades and for satisfactory operation they will not give the miles of service that can be obtained from the heavier grades. For example, an S.A.E. 50 oil of a certain brand will operate on a 3000-mile drain basis, while the same brand in an S.A.E. 30 viscosity in comparable service is unfit for further use after 2000 miles. For clarity I might state that our standard for determining whether an oil is fit for further service is when the oxidized products insoluble in petroleum ether do not exceed 1 per cent. There are other factors to watch, however, but generally, if the sludge content is low, the oil will be fit for further service.

Considerable emphasis has been placed recently by the refiners on the extended service that can be obtained from their various new oils. This recommendation is dangerous unless all of the service requirements are known. Most operators agree today that a good oil could be used almost indefinitely under proper conditions, but therein lies the difficulty. Crankcase conditions and engine operating requirements cannot be controlled generally to sufficiently close limits to make indefinite drain periods practical. Thermostatically controlled engines with adequate oil filters and crankcase ventilation go far in this direction, but the time-temperature relationship must not be overlooked. In checking one of our heavy-duty operations recently, we found that we were using S.A.E. 30 oil of the following specifications:

S.A.E. Viscosity No.	30
Flash Point	440 deg. fahr.
Fire Point	490 deg. fahr.
Pour Point	30 deg. fahr.
Gravity	29.3 deg. A.P.I.
Viscosity, Saybolt at 100 deg. fahr.	419 sec.
Viscosity, Saybolt at 210 deg. fahr.	60 sec.
Viscosity Index	99

The crankcases on these engines held 12 qt., and we had been operating on a 3000-mile drain period. The average truck covered this mileage in five weeks' time. During the period of operation 12 to 14 qt. of oil were added, but the first make-up of one quart did not become necessary until 500 miles had been traveled, and each succeeding quart was necessary at constantly reduced mileage periods of about the following order: 450, 400, 350, and so on—an increase in consumption of from 500 to 100 miles per qt. From this data it would appear that it would be more economical to drain at 2500-mile periods or less. I maintain that the establishment of drain periods is necessary, and that each operation is different, requiring a new laboratory check-up to establish proper time or mileage for draining. No refiner is justified nor any operator warranted in demanding a guaranteed mileage basis for draining until the operation is checked and the requirements very definitely understood.

I follow the practice of keeping a constant running check at several points in the United States where we are using different brands of oil. We select certain hard-working units from which periodic drainings are taken out and sent through for laboratory examination. These oils are then calibrated so we can speak in terms of mileage expectancy. This information is passed on to the field as a guide for draining. If we establish a drain point for hard-working jobs that is satisfactory, we are assured of a good factor of safety for our average operation. Our maintenance expense is approximately 22 per cent of the total expense, not including drivers' salaries. Lubricating oils and greases amount to 3 per cent of this total. Obviously, no operator will gamble very far with attempted reduction in a 3 per cent item which may have a very direct effect on a 22 per cent item.

Table 1—Conventional Engine Oil Specifications
S.A.E. 30

	A	B	C	D	E	F
Gravity, deg. A.P.I.	28.6	25.6	22.0	29.0	23.2	23.1
Flash Point, deg. fahr.	465	445	410	450	400	435
Fire Point, deg. fahr.	525	515	480	510	450	490
Viscosity at 100 deg. fahr.	538	548	640	629	595	508
Viscosity at 210 deg. fahr.	65.7	62	60	71.4	64	58.2
Pour Point, deg. fahr.	—10	5	5	5	20	—10
Conradson Carbon, per cent	0.28	0.32	0.40	0.27	0.0	0.22
Viscosity Index	97	78	41	97	76	62

I do not want to be misunderstood as regards my views on the quality of our present oils. I believe they are the finest we have ever been permitted to work with, and the refiners are to be congratulated on making these improvements at a time when all industry has been confronted with economy programs of varying degrees. My complaint is largely caused by a lack of standardization of both product characteristics and product testing.

Those of us operating large fleets are called upon for specifications by our purchasing division on all products we buy, and later to check to determine if we are receiving these products. We have conventional specifications for lubricating oils, but present-day demands on oils do not permit of service classification under conventional specifications. It is my belief, based on what information is now available, that the two most important characteristics to watch are viscosity index and oxidation susceptibility. In calling for refiners' specifications for comparative purposes we find all sorts of apparent contradictions not as concerns the conventional specifications, but as regards viscosity index and oxidation. Most refiners prefer not to give this information but, when it is extracted, it is in terms of some pet method which their laboratory has evolved and on which there is no general agreement. Before any comparisons can then be made it is necessary to examine the products in our laboratory to correlate the information.

We, the operators, as purchasers are placed in a very peculiar position when we are confronted constantly on this engine oil matter with problems bordering on a research character. I have stated before that operators do not desire to design automotive equipment; neither do we desire to do research work on, or write specifications for, lubricating oils. All that we want and all that we are asking for are definite standards for comparative purposes both on the new oils to be purchased and the old oil drained from service.

The question then arises as to what is a satisfactory crankcase lubricant for commercial use. The answer is obviously as stated before: one that is free-flowing and will eliminate starting difficulties, as well as provide adequate ring seal,

produce the required lubrication and, at the same time, operate with reasonable low oil consumption. I do not believe that operators as a class anticipate that this result can be accomplished with only one of the present S.A.E. classified oils. I believe, further, that the average operator is willing to admit of a separate and distinct summer and winter problem, and is agreeable to making seasonal viscosity changes. Certainly we cannot anticipate that a 20,000-sec. viscosity at 0 deg. fahr. lubricant, which is about the maximum for reasonable starting, can be expected to be suitable for high-temperature work. I do feel, however, that we are definitely headed for two oils, a summer and a winter grade, that will be satisfactory for all of our crankcase lubricating requirements. To accomplish this purpose, however, will require closer cooperation between the engine designer and the oil refiner, as it appears now some of these engines are designed and built first and then turned over to the refiner's lubricating engineers for an oil that will satisfactorily lubricate them. This procedure carries you into the realm of a variety of viscosity oils, and the operator is

Table 2—Crankcase Drainings Fit for Further Service

Mileage	1500	1500	1600	750
Gravity, deg. A.P.I.	27.3	26.0	28.6	27.3
Viscosity, Saybolt at 210 deg. fahr., sec.	72	80	63	80
A.S.T.M. Dilution, per cent	2.80	1.60	2.40	1.20
Insoluble in:				
Petroleum Ether, per cent	0.25	0.73	0.13	0.59
Chloroform, per cent	0.24	0.57	0.08	0.38
Water	None	None	None	None
Ash, per cent	0.24	0.64	0.06	0.32
In Ash: Lead	Mostly	Large	Trace	Mostly
Iron	Small	Small	Trace	Trace

penalized through a lack of standardization. There is, no doubt, a very general feeling on the part of the petroleum industry against obsoleting certain of the high-viscosity oils, however, the trend is definitely in that direction and progress will not be impeded for long.

For comparison I am showing in Table 1 a variety of brands of oil set up on the conventional specification basis. This is the type of information generally supplied to operators and purchasing departments for the procurement of their engine oil requirements. These oils are shown in their true light, stripped of all trade brands, advertising campaigns, and high-pressure salesmanship, and in this light stand or fall on their own merits.

Obviously some of these oils must be accepted, others rejected. When the representatives of the rejected groups are confronted with the reasons for such rejection, invariably we are told that conventional specifications mean little or nothing, and that the only sure way of determining the quality of the oil is to run a practical test. Granting what they state to be the truth, such tests entail considerable expense and time, and it has been my experience that these tests, covering long periods of time, are not satisfactory as it has often developed that during the period of testing the product itself has undergone changes by the refiner in order to overcome certain objections raised, necessitating re-running the tests.

Too much emphasis has been placed in the past on meeting certain conventional specifications, and too little fact has been translated into workable information for the use of the fleet operator.

I touched briefly on the question: When is an oil unfit for further use? There seemingly is less agreement on this question than on the requirements of a new oil. However, to the operator that is the all-important question. We really are not as much concerned with what the specification of the new oil is that goes into the crankcase, as we are with what goes on in the engine during the use of the oil and the time that oil should be removed from the engine to prevent internal damage.

In the last few years we have seen the advent of special-alloy bearings. Although the reasoning behind their use is no doubt sound, new problems have been thrust upon the operator by their introduction. When vehicles operating daily with old-style bearings and doing a satisfactory job with no difficulty, were replaced with units having new-type bearings doing the same type of work, the new units immediately began to give trouble. Manufacturers found many operating reasons why the trouble was caused. However, to the operator these reasons were not apparent until close examination revealed that other elements had been injected into the problem, such as reduction in frontal areas, reduced cooling, and high prevailing crankcase temperatures with a resultant rapid deterioration of the crankcase lubricant. What appeared at first to be a lubricating problem resolved itself into one of engine design and cooling.

We have heard a lot the past year or so about so-called sludge-free oils, and I agree that the sludge problem was becoming rapidly more acute. Having in mind our experiences several years back wherein the lubricants we used seemingly did not have a great deal of sludging tendencies yet, during the past few years with presumably improved lubricants in the same equipment doing the same type of work, we find a constantly increasing amount of sludge formation in the engine. Apparently changes in refinery practice have had something to do with this sludge problem, and the blame does not all rest at the operator's door. The best advice that we received on the problem was to revert from the medium-priced class of lubricants being used to grades in the higher price brackets. Obviously this procedure has been successful as most of us are now using higher-priced oils. For an operator to economically use higher-priced oils he must be able to obtain longer life from them, which resolves itself into stretching the drain period and raises the question of how far these drain periods can be stretched and still be on the safe side from an operating standpoint.

I quote in Tables 2 and 3 the results of laboratory examinations of typical crankcase drainings. I have segregated them into two groups, namely, those which I would consider fit for further service and those which have deteriorated to a point where they would be unsafe for further use. In no case

Table 3—Crankcase Drainings Unfit for Further Service

Mileage	1500	1500
Gravity, deg. A.P.I.	25.1	26.5
Viscosity, Saybolt at 210 deg. fahr., sec.	84	62
A.S.T.M. Dilution, per cent	3.60	8.00
Insoluble in:		
Petroleum Ether, per cent	0.94	1.27
Chloroform, per cent	0.74	1.04
Water	None	None
Ash, per cent	0.93	0.59
In Ash: Iron	Large	Mostly
Lead	Large	Small

Table 4—Analysis of Well-Known Crankcase Oils

	(1) S.A.E 10	(2) S.A.E. 10	(3) S.A.E. 10	(1) S.A.E. 20	(2) S.A.E. 20	(3) S.A.E. 20
Gravity, deg. A.P.I.	32.5	29.0	—	31.6	28.4	—
Flash Point, deg. fahr.	410	385	170	430	410	180
Fire Point, deg. fahr.	455	450	—	475	485	—
Viscosity at 100 deg. fahr., sec.	181	168	115	326	293	250
Viscosity at 130 deg. fahr., sec.	99	91	72	155	141	130
Viscosity at 210 deg. fahr., sec.	49	44	41	59	51	50
Color, N.P.A. No.	4	—	2½-3	5	—	3-3½
Viscosity Index	140	91	—	120	92	—
Carbon Residue, per cent	0.07	0.01	0.15	0.09	0.05	0.7
Pour Point, deg. fahr.	3	0	—20	10	0	—5
Neutralization No.	0.19	0.05	—	0.14	0.03	—
Oxidation		Sligh No. 18 Indiana 20 hr.			Sligh No. 14 Indiana 35 hr.	

	(1) S.A.E. 30	(2) S.A.E. 30	(3) S.A.E. 30	(1) S.A.E. 40	(2) S.A.E. 40	(3) S.A.E. 40
Gravity, deg. A.P.I.	28.7	28.0	26.2	28.8	26.9	—
Flash Point, deg. fahr.	435	445	435	450	460	465
Fire Point, deg. fahr.	480	515	520	510	525	—
Viscosity at 100 deg. fahr., sec.	524	530	541	674	688	720
Viscosity at 130 deg. fahr., sec.	248	233	244	293	294	570
Viscosity at 210 deg. fahr., sec.	65	64	64	74	72	71
Color	N.P.A. No. 5		N.P.A. No. 3½	A.S.T.M. Lt. 5	A.S.T.M. Dk. 4	N.P.A. No. 3½-4
Viscosity Index	99	92	88	101	92	—
Carbon Residue, per cent	0.18	0.19	0.029	0.29	0.33	0.7
Pour Point, deg. fahr.	5	0	0	0/5	0	0
Neutralization No.	0.05	0.03	0.02	0.01	2.54	—
Oxidation		Sligh No. 5	Indiana 70 hr.			
Sludge, per cent after 1 to 5 days				Trace	Trace	Trace
Sludge, per cent after 6 days				Trace	4.5	—
Sludge, per cent after 7 days				Trace	17.0	—

	(4) S.A.E. 40	(5) S.A.E. 40	(6) S.A.E. 40	(1) S.A.E. 50	(2) S.A.E. 50	(3) S.A.E. 50
Gravity, deg. A.P.I.	27.9	26.0	28.7	28.6	26.6	—
Flash Point, deg. fahr.	485	435	440	485	480	500
Fire Point, deg. fahr.	555	495	500	545	540	—
Pour Point, deg. fahr.	0	20	20	0/5	0	10
Viscosity at 100 deg. fahr., sec.	746	656	601	963	1115	1505
Viscosity at 130 deg. fahr., sec.	315	283	270	401	454	570
Viscosity at 210 deg. fahr., sec.	74	71	73	89	92	102.5
Viscosity Index	90	94	109	101	93	—
Color	A.S.T.M. Lt. 5	A.S.T.M. Dk. 6	A.S.T.M. Dk. 5	A.S.T.M. Lt. 6	A.S.T.M. Lt. 5	N.P.A. No. 4-4½
Carbon Residue, per cent	0.27	0.77	0.67	0.30	0.59	1.0
Neutralization No.	5.45	.05	2.9	.01	—	—
Oxidation						70 hr. Indiana
Sludge, per cent after 1 day, S.O.S.[1]	Trace	Trace	Trace	Trace	Trace	
Sludge, per cent after 2 days, S.O.S.	Trace	0.7	Trace	Trace	Trace	
Sludge, per cent after 3 days, S.O.S.	0.5	2.0	Trace	Trace	Trace	
Sludge, per cent after 4 days, S.O.S.	3.0	5.0	Trace	Trace	Trace	
Sludge, per cent after 5 days, S.O.S.	7.0	7.0	Trace	Trace	Trace	
Sludge, per cent after 6 days, S.O.S.	12.0	13.0	Trace	Trace	Trace	
Sludge, per cent after 7 days, S.O.S.	18.0	18.0	Trace	Trace	Trace	

	(5) S.A.E. 50	(6) S.A.E. 50
Gravity, deg. A.P.I.	25.0	28.0
Flash Point, deg. fahr.	440	485
Fire Point, deg. fahr.	510	535
Pour Point, deg. fahr.	15	20
Viscosity at 100 deg. fahr., sec.	1118	1063
Viscosity at 130 deg. fahr., sec.	436	441
Viscosity at 210 deg. fahr., sec.	89	92
Viscosity Index	94	—
Color, A.S.T.M.	Dk. 7	6
Carbon Residue, per cent	1.15	0.69
Neutralization No.	.05	1.94
Oxidation		
Sludge, per cent after 1 day S.O.S	Trace	Trace
Sludge, per cent after 2 days, S.O.S.	Trace	Trace
Sludge, per cent after 3 days, S.O.S.	0.2	Trace
Sludge, per cent after 4 days, S.O.S.	1.5	Trace
Sludge, per cent after 5 days, S.O.S.	5.0	Trace
Sludge, per cent after 6 days, S.O.S.	11.0	Trace
Sludge, per cent after 7 days, S.O.S.	17.0	Trace

[1] Socony oxidation susceptibility.

do we find dilution excessive. In most cases the ash content is low with the exception of those in which the insoluble content is high. This information leads me to the conclusion that the most important factor in determining whether or not an oil is fit for further use is the percentage of insoluble material in the sample. The question then is: What amount of insoluble matter as reflected in the analysis of the crankcase sample is unharmful for further operation?

If, as I have assumed, 1 per cent of insoluble material is the maximum that we can permit for safe operation, then the problem becomes one of selecting proper lubricants that will deliver the desired amount of mileage and lubrication service and remain below the above limit. If it becomes necessary to check each individual operation to determine these facts, then it is a hopeless task for the large operator, and the small operator ordinarily is not equipped to do that sort of checking even if he were so inclined. What we must have

then is some standards set up which have been evolved from a correlation of the laboratory examination of oils with field service. Most refiners know what their oils will do in service as well as their competitors' oils. However, this information obviously cannot be made generally known. I feel, however, that it is only fair to the operators to give them a means of selecting new oils with a full knowledge of what they can expect from them in service.

The data of Table 4 show a laboratory analysis of several brands of viscosities of well-known crankcase oils. It is obvious, if there is a marked difference in these specifications particularly as regards oxidation, that we can expect the same variations in service; but to what extent these variations affect the service obtainable from the oil is not generally known. From a practical standpoint slight variations in the so-called conventional specifications cannot be found in service, due to the wide range of operating variables. However, in checks we have made on various oils, all other factors being reasonably comparable, we find that an oil passing any one of the variety of oxidation tests satisfactorily will give a good account of itself in service. On the other hand, where an oil shows up poorly from the oxidation standpoint, the field results are likewise poor.

Selecting the proper oil to purchase is only one phase of the complicated lubricating-oil problem. The matter of selecting the proper viscosity oils is a very complex one. Where formerly the automobile manufacturers listed their recommendations under summer and winter, or as light, medium, and heavy grades, we now find a growing tendency to give definite recommendations as regards the temperature ranges within which different oils should be used. At first glance you would list them under four classifications, operating temperatures above 90 deg. fahr., between 32 to 90 deg., from 32 to 0 deg., and below 0 deg. This classification seemed to be satisfactory during the period of 1934 and 1935. This year, however, a number of the manufacturers are making as high as six classifications as follows:

No. 1—Above 90 deg. fahr., S.A.E. 50
100 to 30 deg. fahr., S.A.E. 40
65 to 20 deg. fahr., S.A.E. 30
50 to 0 deg. fahr., S.A.E. 20 or 20-W
30 to —15 deg. fahr., S.A.E. 10 or 10-W
—10 deg. fahr. or lower, No. 10-W plus 10 per cent kerosene

No. 2—Above 50 deg. fahr., S.A.E. 30
30 to 80 deg. fahr., S.A.E. 20
10 to 80 deg. fahr., 20-W
—10 to plus 45 deg. fahr., 10-W
—30 to plus 20 deg. fahr., 10-W plus 10 per cent kerosene

No. 3—110 to 70 deg. fahr., S.A.E. 40
110 to 40 deg. fahr., S.A.E. 30
80 to 10 deg. fahr., 20-W
45 to —10 deg. fahr., 10-W
20 to —30 deg. fahr., 10-W plus 10 per cent kerosene

No. 4—Above 90 deg. fahr., S.A.E. 40
Above 40 deg. fahr., S.A.E. 30
85 to 25 deg. fahr., S.A.E. 20 or 20-W
85 to 10 deg. fahr., 20-W
45 to —10 deg. fahr., 10-W
Below —10 deg. fahr., 10-W plus 10 per cent kerosene

No. 5—Above 50 deg. fahr., S.A.E. 30
30 to 80 deg. fahr., S.A.E. 20
10 to 80 deg. fahr., 20-W
—10 to 45 deg. fahr., 10-W
20 deg. fahr. and lower, 10-W plus 10 per cent kerosene

No. 6—110 to 70 deg. fahr., S.A.E. 40
110 to 40 deg. fahr., S.A.E. 30
80 to 10 deg. fahr., 20-W
45 to —10 deg. fahr., 10-W
20 to —30 deg. fahr., 10-W plus 10 per cent kerosene

Among the preceding specifications are three of the most popular cars used in fleet work. Instead of these specifications being worked out to aid the operator in obtaining better efficiency in operation, they have only complicated the situation, particularly in view of the fact that all of us are operating cars of the same make of previous years' models having different lubrication recommendations.

Although I have only taken the time to recite a few of the complications arising from these changes in specifications, there are numerous others equally confusing, and we are farther from a possible standardization than we have ever been up to this time. Does it appear unreasonable on the part of the operators for us to expect better cooperation from the refiners and manufacturers to clarify this problem to enable us to get better results from both of their products?

Transmission and Differential Lubricants

Our problems in connection with the lubrication of transmission and differentials are minor as compared to the engine job. However, we do have lubrication difficulty with these units, much of which is attributed to the fact that research information and field practice are not coordinated. Before we heard of E. P. lubricants in worm-drive axles where contact pressures were high, if we did not get upward of a quarter million miles of service from the worm and wheel, we attributed the ill luck to poor building. However, we must not lose sight of the fact that those axles were built in the days when housings held more than 5 lb. of lubricant. To be called upon to replace a set of transmission gears was an event, and the job was generally carried through to the head of the manufacturer's service department and sometimes to the chief engineer to answer for the failure.

Today we are so accustomed to gear failures necessitating replacements that it has become commonplace. We are in an era now, however, of pint-size transmission cases and gear faces so narrow that slight variations in the meshing of the gears increase tooth pressure to the failure point. As to be expected, when troubles of this character arose, the manufacturer called on the refiner to produce a lubricant that would prevent the welding together of these highly stressed parts in service, and the refiner's answer to this problem has been the so-called E. P. lubricants.

I will not burden you with a review of the reasons for making an E. P. lubricant necessary, and will state further that I subscribe to their use. However, the operator finds himself in a similar situation with these products as confronts him with engine oil, namely, the matter of selecting the proper lubricant to do a satisfactory job.

When it became apparent that many of our difficulties could be overcome through the use of E. P. lubricants, we set about to check the characteristics of the products offered and to determine, in our limited way, the results we could anticipate from their use. Claims were elaborate for the product, but proof was lacking as to what these products would actually do in service. Laboratory tests confirmed our belief that E. P. lubricants could be used to advantage provided the product itself was of a proper type.

In Table 5 is quoted laboratory analysis of typical samples which are reported in the conventional manner. From a chemical aspect there is little to choose one from the other, however, from the physical laboratory standpoint, as well as from field experience, there is a marked difference in the performance of these various products. Some tend to oxidize very readily and become thick and stringy; others separate

Table 5—Analysis of E. P. Lubricants

	(13) S.A.E. 90	(13) S.A.E. 110	(2) S.A.E. 90	(2) S.A.E. 160
Gravity, deg. A.P.I.	22.0	22.0	25.6	24.0
Viscosity, S.U.V. at 100 deg. fahr., sec.	1100	3800	1095	3758
Viscosity, S.U.V. at 130 deg. fahr., sec.	455	1200	457	1359
Viscosity, S.U.V. at 210 deg. fahr., sec.	100	160	97	207
Viscosity, Saybolt Furol at 100 deg. fahr., sec.	115	385	111	381
Viscosity Index	100	80	102	102
Flash Point, Open Cup, deg. fahr.	450	520	475	580
Fire Point, Open Cup, deg. fahr.	515	575	545	655
Pour Point, deg. fahr.	—5	10	0	5
Color, N.P.A. No.	6 Dil.	7 Dil.	5.5 Dil.	7.5 Dil.
Sulphur, per cent	0.6	0.6	1.50	1.55
Chlorine, per cent	0.5	0.5	—	—
Corrosion Test at 212 deg. fahr.	Slight Tarnish	Slight Tarnish	Slight Tarnish	Slight Tarnish
Load Capacity:				
Almen Rating at 600 R.P.M.	30	30	20	18
Lever Load, lb.	Torque 30	Torque 30	—	—
Timken Rating at 800 R.P.M.				
100 deg. fahr., 10 min., in lb.	33	33	50	45
S.A.E. Transmission and Rear-Axle Viscosity No.	90	110	90	160

rapidly. Some lubricants have a tendency to foam excessively, producing high level under operating conditions and aggravating the already difficult brake problem.

In our studies of E. P. lubricants we have been extremely fortunate in having accessible a Timken, Almen, Floyd-Hydraulic, Faville-Levally, and new S.A.E. Standard machine for physical test purposes. I am showing below test data obtained by using the same lubricant on all four machines. Obviously no correlation is readily possible.

The test sample used for these runs was an S.A.E. 110 E. P. lubricant, which chemically is a sulpho-chlorinated fatty oil base compounded with 0-deg. cold test mid-continent bright stock. Under the standard Almen test at 600 r.p.m. the following data were obtained:

2-lb. Weights	Torque, ft. lb.
1	1.5
2	6.0
3	8.5
4	11.0
5	13.5
6	16.5
7	20.0
8	23.0
9	22.5
10	22.0
11	23.0
12	23.5
13	24.0
14	27.0
15	34.0

This sample went the full limit of the Almen test without any indications of scuffing or seizure.

Using the same sample on the Faville-Levally machine, at 388 r.p.m.:

Load, lb.	Torque, in. lb.
250	3
500	11
750	19
1000	31
1250	35
1500	39
1750	35
2000	34.5
2250	37.0
2500	37.5
2750	39.0
3000	41.0
3250	43.0
3500	43.5
3750	45.0
4000	45.5
4250	46.0
4500	47.0

This sample likewise reached the limit of this machine without seizure, breaking the driving pin on account of high torque. Test blocks indicated excessive scoring but no seizure. Computing the area of contacts with a total load of 4500 lb. on the machine, the lubricant was subjected to approximately 66,000 lb. per sq. in. on the rubbing surfaces between the pin and blocks.

The same sample tested on the hydraulic-type Floyd machine operating at 100 r.p.m. went the full limit of the capacity of the machine, which is 600 lb. per sq. in. gage reading, without failure of any character. A slight swedging action was noticed upon inspection of the pin.

The same sample tested on the Timken machine at 800

Table 6—Recommended Specifications of Typical Wheel-Bearing Greases

	Soap Content (Maximum) Per Cent	Water (Maximum) Per Cent	A.S.T.M. Worked Penetration	Dropping Point (Minimum) (Deg. Fahr.)	Texture	Saybolt Viscosity of Oil Content (Minimum) Sec.
Calcium Base	18	1	185-215	200	Smooth	480 at 100 deg. fahr.
Sodium Base	13	0.5	280-310	325	Fibrous	130 at 210 deg. fahr.

Table 7—Analysis of Wheel-Bearing Grease

	(7)	(8)	(9)	(10)
Ash, per cent	1.20	2.81	1.79	1.29
Calcium	Heavy		Very Heavy	Absent
Magnesia	Trace		Trace	Absent
Aluminum	None		Heavy	Absent
Iron	Present		Heavy	Trace
Potassium and Sodium	Trace		Present	Trace
Insoluble in Ash, per cent	0.06	None	0.76	1.03
Solubility in Kerosene	Clear		Clear	Cloudy
Moisture, per cent	1.05	0.71	0.72	0.13
Free Alkali	None	None	None	None
Soap (As Calcium Oleate), per cent	10.54	8.42	0.30	4.26
Mineral Oil (By Difference), per cent	88.35		98.22	94.52
Consistency at 100 deg. fahr.	Easy to Stir	Almost Solid. Thick & Hard to Stir	Gelatinous Thinner Than at Room Temperature	No Noticeable Change from Room Temperature
Consistency at — 10 deg. fahr.	Easy to Stir	Solid	Cannot Stir Stiffer Than at Room Temperature	Very Stiff Not Solid

r.p.m. went the full 33 lb. on the beam without failure. With a total load of 330 lb. and an approximate scar area of .015 sq. in., the lubricant was subjected to an approximate pressure of 22,000 lb. per sq. in.

Using the same sample on the S.A.E. Standard machine with the following setup: top spindle speed 731 r.p.m., ratio of top spindle to lower spindle 10.4 to 1, loading speed, 26.3 lb. per sec., the sample showed consistently better than a 200-lb. beam load at time of seizure.

The Standard S.A.E. machine does not offer any ready means of measuring the torque and, no doubt, many refinements will be required before this machine is adaptable for general all-around use. This development will require time and, in the meantime, the buyer will be furnished specifications of gear oils with their load-carrying ability furnished from any of the previously mentioned machines.

In view of the time element involved it seems to me that some tentative standards could be set up covering the various machines now in general use, these standards later to be supplemented with correlating data obtained from the S.A.E. Standard machine.

I appreciate the work that is being done by the Research Committee, Lubricants Division, of the Society in connection with the testing of E.P. lubricants, however, operators are not in sympathy with a fact-finding body's progress being retarded on account of an irrelevant difference of opinion. It is imperative that we have a standardized procedure for determining the quality of these products.

Those of us who have standardized on E.P. lubricants face a situation that is troublesome in the purchase and placing in service of new equipment. The majority of the manufacturers are not using E.P. lubricants in the original filling of transmissions and differentials. When this equipment goes into service we are either faced with the removal of the original lubricants immediately and refilling with the E.P. lubricants, which we have in stock, or with running the risk of having E.P. lubricants added to other types, which in my experience is likely to cause trouble.

Many of the manufacturers are non-committal in their specifications regarding E.P. lubricants; yet some that do not use them in their factory raise no objection to the use of these lubricants in their products.

I have been using a sulpho-chlorinated fatty oil, compounded with a 0-deg. cold test mid-continent bright stock E.P. lubricant over a period of four years with excellent results. The only difficulty encountered was in the transition period of changing from soap-filled lubricants to those of the E.P. type. Irrespective of the care exercised in thoroughly cleaning housings before application of the new type of lubricant, it was found that the felt retainers were so filled with soap that they would not do a satisfactory sealing job and, until new retainers were installed, considerable difficulty was encountered with leaks. Jobs that were reporting leaks

Table 8—Recommended Specifications of a Satisfactory Water-Pump Grease

Calcium Soap Base (Maximum) Per Cent	Water (Maximum) Per Cent	A.S.T.M. Worked Penetration	Dropping Point (Minimum) Deg. Fahr.	Saybolt Viscosity of Oil Content (Minimum) Sec.
24	1	160-190	210	100 at 100 deg. fahr.

Table 9—Recommended Specifications of Semi-Fluid Pressure-Gun Greases

	Soap Content (Maximum) Per Cent	Water (Maximum) Per Cent	A.S.T.M. Worked Penetration	Texture	Saybolt Viscosity of Oil Content (Minimum) Sec.
Calcium Base	6 Cal.	1	350-400	Smooth	140 at 210 deg. fahr.

Table 10—Analysis of Chassis Gun Grease

	(15)	(16)
Ash, per cent	1.99	1.16
Insoluble in Ash, per cent	0 09	0.03
Solubility in Kerosene	Cloudy	Soluble
Moisture, per cent	0.23	3.90
Free Alkali	None	None
Soap (As Calcium Oleate), per cent	9.49	0.12
Mineral Oil (By Difference), per cent	90.19	95.95
Consistency at 100 deg. fahr.	Same as Room Temperature	Soft, Easily Stirred
Consistency at —10 deg. fahr.	Stiffer than Room Temp.	Stiff, Easily Stirred

Table 11—Manufacturers' Specifications for Six Passenger Cars

Car	Year Model	Crankcase S.A.E. and W. Grades: Above 90 deg.	90 deg. to 32 deg.	32 deg. to 0 deg.	0 deg. to -15 deg.	Transmission S.A.E.: Over 32 deg. Summer	Below 32 deg. Winter	Differential S.A.E.: Over 32 deg. Summer	Below 32 deg. Winter	Steering	Chassis	Water Pump
Car A	'31	50	40	20-W	10-W	160	90	160 E.P.	90 E.P.	Gear Lube	Chassis Lube	Pump Grease
	'32	50	40	20-W	10-W	160	90	160 E.P.	90 E.P.	" "	" "	Chassis Lube
	'33	50	40	20-W	10-W	160	90	160 E.P.	90 E.P.	" "	" "	" "
	'34	50	40	20-W	10-W	160	90	160 E.P.	90 E.P.	" "	" "	" "
	'35	50	40	20-W	10-W	160	90	160 E.P.	90 E.P.	" "	" "	" "
	'36	50	40	20-W	10-W	160	90	160 E.P.	90 E.P.	" "	" "	" "
Car B	'31	30	30	20-W	10-W	160	90	160	90	Gear Lube	Chassis Lube	Pump Grease and Oil
	'32	30	30	20-W	10-W	160	90	160	90	" "	" "	" " " "
	'33	30	30	20-W	10-W	160	90	160	90	" "	" "	" " " "
	'34	30	30	20-W	10-W	160	90	160	90	Non-Fluid Lube	" "	" " " "
	'35	20	20-W	10-W	10-W	160	90	160	90	" "	" "	" " " "
	'36	30	20	10-W	10-W	160	90	160	90	St'g Gear Lube	" "	" Engine Oil
Car C	'31	40	30	20-W	10-W	60	40	160	90	St'g Gear Lube	Semi-Fluid Lube	Pump Grease and Oil
	'32	40	30	20-W	10-W	160	90	160	90	" " "	" "	" " " "
	'33	40	30	20-W	10-W	160	90	160	90	Fluid Gear Lube	" "	Pump Grease
	'34	40	30	20-W	10-W	160	90	160	90	" " "	" "	" "
	'35	40	30	20-W	10-W	160	90	160	90	" " "	" "	" "
	'36	30	20-W	10-W	10-W	160	90	160	90	" " "	" "	" "
Car D	31	40	30	20-W	10-W	60	40	160	90	Fluid Gear Lube	Semi-Fluid Lube	Pump Grease and Oil
	'32	40	30	20-W	10-W	160	90	160	90	" " "	" "	" " "
	'33	40	30	20-W	10-W	160	90	160	90	" " "	" "	Pump Grease
	'34	40	30	20-W	10-W	160	90	160	90	" " "	" "	" "
	'35	40	30	20-W	10-W	160	90	160	90	" " "	" "	" "
	'36	30	20-W	10-W	10-W	160	90	160	90	" " "	" "	" "
Car E	'31	40	30	20-W	10-W	160	90	160	90	Fluid Gear Lube	Chassis Lube	Engine Oil
	'32	40	30	20-W	10-W	90	80	160	90	" " "	" "	" "
	'35	30	20	10-W	10-W	160	90	160	90	Spec. " "	" "	" "
	'36	30	20	10-W	10-W	160	90	160	90	St'g Gear Lube	" "	Light Mach. Oil
Car F	'34	30	30	20-W	10-W	160	80	160	90	St'g Gear Lube	Chassis Lube	Engine Oil
	'35	30	30	20-W	10-W	160	80	160	90	" " "	" "	" "
	'36	30	20	10-W	10-W	160	90	160	90	" " "	" "	" "

using S.A.E. 250, E.P. lubricants, after installation of new seals operated satisfactorily without leaks on S.A.E. 110 E.P. lubricants during summer operation.

Chassis Lubrication

Many improvements have been noted in chassis lubricating greases, and we on the operating side are favorably impressed by the tendency of these greases to have extreme-pressure characteristics, enabling us to service a wider variety of equipment satisfactorily with a lesser number of lubricants. We are still very much concerned, however, with the manufacturer's specifications, as to what he feels is required in the way of lubricants to do a satisfactory job on his particular chassis, and it further complicates the matter when, during his production year, lubrication changes are made in the specifications, although the instruction manual furnished with current production does not show these changes.

For example, one manufacturer last year had two entirely different types of water pumps on one model. One pump had a spring-loaded packing and was intended to be lubricated with chassis lubricant; special water-pump grease was not recommended and would not give proper satisfaction. The newer-type pump, used on part of the production, was lubricated with engine oil through an oil cup, and the factory recommendation for it was an S.A.E. 40 oil for summer and an S.A.E. 20 oil for winter. It is expected, of course, that the night oiler in the garages will not replace the oil cup with a grease nipple to suit his convenience, but that he will discover that two trucks of the same make and model that may appear identical require different lubricants for the water pump. Considerable confusion exists anyway regarding the lubrication of water pumps and fan bearings, and a great deal of service complaints could be eliminated if this troublesome point were cleared up.

In the case of one manufacturer of trucks that I call to mind, on two models the rear-wheel bearings are lubricated by removing the wheels, but on three other models the rear-wheel bearings are provided with a grease fitting so that lubricant can be applied with a gun instead of by removing the wheels. It would simplify the lubrication problem and assure the manufacturer better service if all of these models were provided with fittings to lubricate without the removal of rear wheels. Present-day design is making the chassis extremely difficult to lubricate—art has won the battle over practicability. We must remove a wheel or a fender to adjust valve lash and, to drain oil or drain the transmission, it is

imperative that you have either a pit or a lift. Some of our operations, unfortunately, do not warrant such equipment. To find the grease fittings requires a contortionist. Chassis designers seemingly have little knowledge of how many grease fittings will be inaccessible after the body is mounted, apparently until the jobs start coming off at the end of the line. Designing for accessibility is a lost art.

We attempt to limit the number of lubricants carried in our garages to the following: two viscosities of crankcase oils, special water-pump grease, wheel-bearing grease that is suitable also for grease-type universal joints, transmission and differential lubricant, and high-pressure-gun grease.

If you follow the manufacturers' recommendations steering gears should come in for a special lubricant. We find that the E.P. lubricant we use in transmissions and differentials does a satisfactory job here.

Universal-joint lubrication is very important. This joint is unhandy to reach, often neglected, and the hardest working point to come under chassis lubrication. The compromise on one lubricant for front-wheel bearings and grease-type universals is not a happy one, due primarily to the lack of ability of the average pressure gun to handle a grease of this consistency. The oil-type joint is easier to service, and some of the later designs are stated to be packed for the life of the joint. However, if this be true, the life will be relatively short, as we find it necessary to re-lubricate these joints at frequent intervals.

Shackle lubrication is a constant problem. We have been confronted with, and still are operating, many different types of shackle construction, including the straight-bolt type, roller-bearing type, threaded type, and later the U type with threads on both arms. In the lighter equipment we notice a trend toward the rubber-type non-lubricated shackle. It has been found impossible to lubricate some of these types of shackles with any lubricant, which required their replacement with a type that was more satisfactory. We find that with this variety of types of shackles, we get along very well with a semi-fluid pressure-gun grease or a regular E.P. transmission lubricant. The disadvantage of using the fluid type is a loss of oil from the gun and the mess it makes around the grease rack, as well as on the chassis.

I am showing for comparison in Tables 6 to 10 inclusive only brief specifications of typical greases, along with some of our laboratory findings of samples analyzed.

In analyzing manufacturers' specifications for lubricants for the six makes of passenger cars generally used in fleet work (see Table 11), we find for year 'round service that we would require the following:

Above 90 deg. fahr. requires four viscosity grades of engine oil.

Between 90 and 32 deg. fahr. requires four viscosity grades of engine oil.

From 32 to 0 deg. fahr. requires two viscosity grades of engine oil.

From 0 to —15 deg. fahr. requires one viscosity grade of engine oil.

Transmission, summer requires three grades—winter, three grades.

Differential, summer, two grades—winter, two grades.

They do not offer specific recommendations for general

Table 12—Manufacturers' Specifications for Four 1½-Ton Trucks

Truck	Year Model	Crankcase S.A.E. and W. Grades: Above 90 deg.	90 deg. to 32 deg.	32 deg. to 0 deg.	0 deg. to -15 deg.	Transmission S.A.E.: Over 32 deg. Summer	Below 32 deg. Winter	Differential S.A.E.: Over 32 deg. Summer	Below 32 deg. Winter	Steering	Chassis	Water Pump
Truck B	'31	30	30	20-W	10-W	160	90	160	90	St'g Gear Lube	Chassis Lube	Water-Proof
	'32	30	30	20-W	10-W	160	90	160	90	" " "	" "	" "
	'33	30	30	20-W	10-W	160	90	160	90	" " "	" "	" "
	'34	30	30	20-W	10-W	160	90	160	90	" " "	" "	" "
	'35	30	20	10-W	10-W	160	90	160	90	" " "	" "	" "
	'36	30	20-W	10-W	10-W	160	90	160	90	" " "	" "	" "
Truck C	'31	40	30	20-W	20-W	160 E.P.	90 E.P.	160 E.P.	90 E.P.	S.A.E. 160 90-W	Semi-Fluid Lube	Pump Grease
	'32	40	30	20-W	20-W	160 E.P.	90 E.P.	160 E.P.	90 E.P.	" "	" "	" "
	'33	40	30	20-W	20-W	160 E.P.	90 E.P.	160 E.P.	90 E.P.	" "	" "	" "
	'34	40	30	20-W	20-W	160 E.P.	90 E.P.	160 E.P.	90 E.P.	" "	" "	" "
	'35	40	30	20-W	20-W	160 E.P.	90 E.P.	160 E.P.	90 E.P.	" "	" "	" "
	'36	40	20	20-W	10-W	160 E.P.	90 E.P.	160 E.P.	90 E.P.	" "	" "	" "
Truck A	'31	50	40	20-W	10-W	160	90	160	90	Gear Lube	Chassis Lube	Chassis Lube
	'32	50	40	20-W	10-W	160	90	160 E.P.	90 E.P.	" "	" "	" "
	'33	50	40	20-W	10-W	160	90	160 E.P.	90 E.P.	" "	" "	" "
	'34	50	40	20-W	10-W	160	90	160 E.P.	90 E.P.	" "	" "	" "
	'35	50	40	20-W	10-W	160	90	160 E.P.	90 E.P.	" "	" "	" "
	'36	50	40	20-W	10-W	160	90	160 E.P.	90 E.P.	" "	" "	" "
Truck G	'31	50	40	20-W	20-W	160	90	160	90	Gear Lube	Chassis Lube	Pump Grease
	'32	50	40	20-W	20-W	160	90	160	90	" "	" "	" "
	'33	50	40	20-W	20-W	160	90	160	90	" "	" "	" "
	'34	50	40	20-W	20-W	160	90	160	90	" "	" "	" "
	'35	40	30	20-W	10-W	160	90	160	90	" "	" "	" "
	'36	40	30	20-W	10-W	160	90	160	90	" "	" "	" "

Table 13—Comparison Manufacturer's Specifications with Refiners' Recommendations

		(A) Manufacturer	(2) Refiner	(4) Refiner	(3) Refiner
Crankcase	Summer	Viscosity, Saybolt, Sec. 1000 Max. at 100 deg. fahr. 65 Min. at 210 deg. fahr.	New, S.A.E. 30 After 3,000 Miles, 40 After 25,000 Miles, 50	Above 90 deg. fahr., 40 90 to 32 deg. fahr., 30	40
	Winter	400 Max. at 100 deg. fahr. 50 Min. at 210 deg. fahr.		32 to 0 deg. fahr., 20-W 0 to −15 deg. fahr., 10-W	20
Transmission	Summer	190–120 at 210 deg. fahr.	160 E.P.	160 E.P.	160
	Winter	The above with 10 per cent Kerosene	North, 90 E.P. South, 110 E.P.	90 E.P.	90
Differential	Summer	1200/1600 at 130 deg. fahr. 200/230 at 210 deg. fahr.	See Transmission Specifications	See Transmission Specifications	160 E.P.
	Winter	860 Max. at 130 deg. fahr. 135/145 at 100 deg. fahr.			90 E.P.
Chassis		Pressure Grease Viscosity of Mineral Oil 275/300 at 100 deg. fahr.	Chassis Lubricant	Chassis Lubricant, Viscosity of Mineral Oil 750 at 100 deg. fahr.	Chassis Lubricant
Steering		Same as Transmission	Same as Transmission	Same as Transmission	Same as Transmission
Water Pump		No. 4 Cup Grease Min. 75 at 210 deg. fahr.	Chassis Lubricant	Chassis Lubricant 750 at 100 deg. fahr.	Chassis Lubricant
Universal Joints		Universal Grease, Viscosity 85/90 at 210 deg. fahr.	Universal Grease	Universal Grease, Viscosity 170 at 210 deg. fahr.	Universal-Joint Lubricant
Front Wheels		Short-Fiber Grease, Viscosity 290/310 at 100 deg. fahr.	Universal Grease	Wheel-Bearing Grease	Wheel-Bearing Grease

Table 14—Comparison Manufacturer's Specifications with Refiners' Recommendations

		(B) Manufacturer	(2) Refiner	(4) Refiner	(3) Refiner
Crankcase	Summer	Over 50 deg. fahr., 30 30 to 80 deg. fahr., 20	New Engine, 20 or 20-W After 3,000 Miles, 20 or 20-W After 25,000 Miles, 30	Above 90 deg. fahr., Trk. 30 Pas. 20 90 to 32 deg. fahr., 20	Truck, 40 Car, 30
	Winter	−10 to 45 deg. fahr., 10-W Below 20 deg. fahr., 10-W with 10 per cent Kerosene		32 to 0 deg. fahr., 10-W 0 to −15 deg. fahr., 10-W	Truck, 30 Car, 20
Transmission	Summer	160	160 E.P.	160 E.P.	160
	Winter	90, Below 0 deg. fahr. Add 10 per cent Kerosene	North, 110 E.P. South, 90 E.P.	90 E.P.	90
Differential	Summer Winter	Same as Transmission	Same as Transmission	Same as Transmission	Same as Transmission
Chassis		Chassis Lubricant	Chassis Lubricant	Chassis Lubricant	Chassis Lubricant
Steering		Steering-Gear Lubricant	Chassis Lubricant	Same as Transmission	Same as Transmission
Water Pump		Water-Proof Grease Fill Cup with Engine Oil	Water-Pump Lubricant, 40	Water-Pump Lubricant	Water-Proof Grease
Universal Joint	Front	From Transmission	From Transmission	From Transmission	From Transmission
	Rear	Transmission Grease	Universal Grease	160	160
Front Wheels		High-Melting-Point, Front-Wheel-Bearing Grease	Universal Grease	Wheel-Bearing Grease	Wheel-Bearing Grease

Table 15—Comparison of Six Popular Fleet Cars and Four 1½-Ton Trucks

Cars	Make	Number of Lubrication Points	Number of Different Lubricants Required
	A	30	7
	B	34	8
	C	40	6
	D	36	6
	E	35	6
	F	44	6
Trucks	A	45	6
	B	42	8
	C	46	6
	G	53	6

chassis lubrication other than such captions as "Fluid Gear Lubricant," "Chassis Lubricant," "Water Pump Grease," and so on.

In analyzing manufacturers' specifications for four popular makes of 1½-ton trucks in Table 12 we find for crankcase lubricants:

Above 90 deg. fahr., four viscosity grades of engine oil.

Between 90 and 32 deg. fahr. requires four viscosity grades of engine oil.

From 32 to 0 deg. fahr. requires two grades.

From 0 to —15 deg. fahr. requires two viscosity grades of engine oil.

For year 'round operation we use a total of six grades of crankcase oil and, on transmission and differentials, we find it will require four grades for year 'round operation.

In conclusion, I feel that the W grades of oil have justified their position in the classification of engine oils and that the advantages obtained from their use in winter operation offset their slightly increased oil cost due to increased consumption.

Field experience and laboratory research are in accord as regards the practicability of using lower viscosity oils than are now generally used for higher temperature work. It is hoped that this information will be more generally used by designing engineers in building engines of the future, enabling us to do a wider variety of work with fewer crankcase oils.

A definite need has been established by the operators for standardized methods of testing new engine oils for determining their suitability for service, particularly as regards their susceptibility to oxidation.

As with new oils, so with used oils is it imperative that we establish standards for determining when an oil is unfit for further service.

It is hoped a better understanding will prevail between manufacturers and refiners as regards the characteristics of E.P. lubricants, permitting a better standardization and a more general acceptance of these highly desirable products in the lubricant line.

Likewise, there is a very definite need for a standardized procedure of testing to enable the proper selection of E.P. lubricants. There is a feeling of skepticism on the part of the majority of operators regarding E.P. lubricants, and I think justly so. Until such a time as better information is available on the subject this market will be stifled. It is expected of the Society that it will use its facilities for determining the scientific facts in the case, rather than permitting the issues to become fogged because of the commercial angles of the problem.

One of the most serious complaints registered by the commercial operator against the automotive industry is the difficulty in lubricating the average chassis. Inaccessibility of fittings is paramount. Untold waste could be eliminated annually from fleet operation if improvements were made in this direction. It is hoped the designing engineer will give more thought to this problem rather than delegating that important phase of the work to drafting departments. We, on the operating side, are not unmindful of the many problems that both the manufacturer and the refiners are confronted with in merchandising their products. However, we feel that if many of the things we are requesting are put into practice, their service burdens at least will be lightened.

I wish to take this opportunity to thank those of the refiners and manufacturers who cooperated in furnishing their lubricating recommendations for various makes and models of cars and trucks. Also, those of the Transportation and Maintenance Committee that returned their questionnaire on their field practice. I wish to mention especially Standard Oil Co. of Indiana, Standard Oil Co. of New York, Standard Oil Co. of New Jersey, Atlantic Refining Co., Gulf Refining Co., Sinclair Refining Co., Pure Oil Co., D. A. Stuart Co., R. R. Mathews of the Battenfield Oil and Grease Corp., and the Chek-Chart Corp.

Discussion

Several Lubricants Necessary for Efficient Operation

—Leonard V. Newton

Byllesby Engineering and Management Corp.

IN our motor-vehicle fleet operations, the cost of lubricants represents a very small percentage of our total operating costs. The labor involved in changing oil, checking oil levels, and lubricating chassis is quite considerable. The potential cost of replacement parts and labor of installing them, which might be caused by use of wrong lubricants or lack of lubrication, is great.

In our consideration of this subject, I should like to stress the thought that our primary aim should be to select and use those lubricants that will promote long life of wearing parts due to their ability to reduce friction, thereby not only minimizing wear, but also increasing performance.

To accomplish this purpose, I believe that we will have to continue using several grades of motor oil, transmission and rear-axle lubricants, and specialized lubricants for specific chassis parts. While we, as operators, want to reduce the number of lubricants to a minimum, we must not lose sight of the real purpose of lubrication. Our job of lubricating motor vehicles is not any more complex than that which plant engineers face in their problem of lubricating specialized machinery—they still use many types and grades of lubricants.

Regarding oil viscosity, I agree with Mr. Faulkner that we want to use light motor oils. Where high-viscosity oils have been or are still used, it is due either to the operator's desire to economize foolishly on oil or to the specific recommendation of the manufacturer.

We have always had winter problems in our motor-vehicle operations because, in the public utility business, increased vehicle usage occurs during extremely cold weather and dur-

ing bad storms. The oil companies contacting us have always recommended the lighter viscosity, 10-W pour test oils for cold-weather use. I have noticed that many operators who are influenced by apparent oil economy persist in using heavier oil than necessary because they consume less oil. Oil companies have not dared to be over emphatic in their suggestions regarding light oil for fear that the higher oil consumption would jeopardize their business.

Rather than a compromise to hold oil consumption to a definite small loss by the recommendation of an oil with pour point seriously affecting starting, I would recommend 0-deg. pour test oils and grade them on their merits to lubricate correctly. The oil I choose may be consumed more rapidly, but it will lubricate the engine. Not so long ago we saw a practical open-throttle test in which it was proved conclusively that, of several oils used in duplicate tests, greater speed was obtained during the periods in which light oils were used.

Given oils of a certain viscosity at either 100 deg. or 210 deg. fahr., I find that comparative consumption is somewhat influenced by boiling points and volatility, preference being given the oil showing the smallest volatility loss and highest initial boiling point.

In our garages we find it practical to carry several grades of motor oils, from 10-W to S.A.E. 40, and specialized lubricants to lubricate passenger cars and trucks properly. Passenger cars range in age from brand new to 5 yr., trucks from new to 14 yr. and, in capacity, from ½ to 7½ tons.

Bear in mind that we continue to maintain old equipment for the sake of economy, and the characteristics of design in old vehicles is far different than those in current vehicles.

The oil refiner cannot provide us with a single lubricant that will lubricate correctly all makes and types of vehicles. It is up to us to provide all types and grades of lubricants to afford correct lubrication.

I was interested in Mr. Faulkner's observation that the use of S.A.E. 40 oil in a new vehicle during break-in periods causes high consumption of S.A.E. 20 when subsequently used. This observation proves that S.A.E. 40 oil caused wear and did not provide adequate lubrication when the engine was tight. I imagine that the S.A.E. 40 oil when drained was of greater specific gravity and of better appearance than the S.A.E. 20 oil. If the latter had been used originally, the engine might have been properly broken in and abnormal oil consumption might not have resulted.

I subscribe to the use of an oil of a viscosity that will permit the engine to develop its maximum power. Too light an oil will not provide sufficient film strength to prevent blowby —too heavy an oil causes oil drag.

When Mr. Faulkner refers to the new light oils, does he refer to oils produced by new refining processes? Many light oils are of the new selective-solvent-extraction type; others are produced by more conventional refining methods, such as fractional distillation and subsequent treatment and filtration.

I believe that the type of oil as well as the time element in service has a distinct bearing on sludging.

Few refiners recommend the use of their oils for long periods, the maximum regularly advocated being 2000 miles, and that by only one manufacturer. Long periods between drains were fostered by car manufacturers about the time oil filters and crankcase ventilation were introduced.

Any motor oil becomes contaminated with blowby fuel, which may vary greatly in degree at different seasons of the year. Such contamination over prolonged periods of time has a direct bearing on sludging. The type of fuel used also has its effect.

Mr. Faulkner's record of extreme change between the 500-mile period and the time of final draining places the drain period even below 2500 miles. It would be interesting to check the oils for lubricating quality by measuring power output over the same periods.

Mr. Faulkner's suggestion for greater cooperation between engine builders and oil companies is very important. Refiners will produce any lubricant required and, generally speaking, one good refiner can duplicate another's product. Engine builders are not in accord in their ideas on engine design, neither are some of their theories of lubrication in accord. Closer cooperation between engine designers and builders might help materially.

Specifications of Motor Oils Used

	10-W	20-W	S.A.E. 20	S.A.E. 30	S.A.E. 40
Gravity, deg. A.P.I.	27.0	26.5	25.5	25.5	24.8
Flash-Fire Points, deg. fahr.	400-460	415-480	435-500	445-510	460-525
Viscosity at 100 deg. fahr., sec.	180	240	400	505	750
Viscosity at 210 deg. fahr., sec.	45	48	55.5	60	70
Pour Test, deg. fahr.	0—	0—	0—	0—	0—
Viscosity Index	94	91	81	76	73
Conradson Carbon, per cent	0.064	0.09	0.23	0.365	0.59
Initial Boiling Point, deg. fahr.	740	752	760 plus	760 plus	760 plus

I notice that Mr. Faulkner attributes difficulty in lubricating new types of bearings to engine design, further stating that no doubt the reasoning establishing the new types is sound. The question is, are the new bearings sound in design or is this simply a case of divergence in opinion of engine designers? Many continue the older types of bearings, and few use exactly the same kind and type.

Regarding sludging, we are all more or less sludge minded. Advertising campaigns have been built around the subject. Mr. Faulkner states that it was non-existent a few years ago. I believe that, if it exists today, it existed then. Oils have not gone backwards, I am sure; in fact, one great factor in sludging effect has been recommended in all types of oil (solvent or conventional type), and that is wax. At one time amorphous wax settled out in crankcases, but that was before the day of centrifuges and the use of propane.

I do not agree with Mr. Faulkner's statement that, for an operator to use higher priced oils economically, he must be able to obtain longer life, but I do believe that he must obtain better lubrication, better engine performance, and greater engine protection against wear.

Mr. Faulkner's comments on insolubles are interesting. The analysis to determine the nature of the insolubles is important and most likely it will vary greatly for each sample tested.

Good oil filters, properly serviced, should reduce materially the percentage of insolubles.

Oxidation tests have possibilities. Selective solvent oils should have an oxidation number of 0. There is controversy on the method of determining oxidation, the Bureau of Standards, Standard Oil of Indiana, Sligh, and other methods being used. Is make-up oil included in oils analyzed in Table 4 of the paper? What would the analysis of these oils show after 1000 miles of service?

I agree that manufacturers' recommendations are complex when they recommend 6 grades of oils including kerosene-diluted 10-W oil. The number of lubricants can be reduced if vehicles are housed in heated garages and properly protected for winter operation with radiator protectors, hood shields, thermostats in water outlets, and so on.

Regarding E. P. lubricants, we have available soap-filled E. P. lubricants that may be obtained either in grease or oil consistency. We prefer the oil consistency whenever it may be employed. I am in accord with the statement that E. P. lubricants are necessary; our oil companies are now working on this subject. Whether the sulphurized or chlorinated type or combination of both is best is still questionable. As far as testing machines are concerned, there is also a question as to whether testing machines should work to the seizure point, which condition is really not duplicated in actual usage, or to alleviation of friction.

Regarding troubles brought about by adding make-up oil to original housing fill, many operators have complained of them. We use only one make and type and eliminate this difficulty. The trouble seems to be caused either by the fact that dissimilar chemicals will not mix or that the proper balance is disturbed. Refiners must sooner or later meet this problem by identifying the drum or package of E. P. lubricants by showing the chemical type.

Steering-gear lubrication requires a special lubricant; its consistency must be such as to lubricate without retarding the flexible operation of the gear. Some transmission oils would cause gears to be stiff. At this point I would like to stress the necessity for a drain plug in steering-gear housings.

Referring to universal-joint greases, in Spicer-type joints, the important thing is to employ a grease that will resist centrifugal force and not throw away from bearing surfaces.

Of wheel-bearing greases, we have two types, smooth and short fiber. We are using the smooth calcium-type lubricant and are having no difficulty during cold weather. I have found no efficient wheel-bearing grease that can be used for lubrication of grease-type universal joints.

For proper shackle lubrication we require a grease of clinging consistency, water and heat resistant.

For transmission and rear-axle lubrication we are using two types of transmission oil, S.A.E. 110 and S.A.E. 160, both of E. P. type. In some few vehicles with free-wheeling units we use S.A.E. 90 transmission oil.

In conclusion I would like to state that, following progress in the automotive world, lubrication recommendations and procedure are being altered constantly by the vehicle manufacturers as a result of changes in production or because of service experience in the field.

No mention has been made in Mr. Faulkner's paper of break-in oils, either graphital or otherwise, nor has mention been made of oils to be added to conventional motor oil to increase film strength.

We add about 1 qt. of light motor oil to each 5 to 10 gal. of fuel during the first 1000 or 1500 miles of a vehicle's life. I believe this scheme has considerable merit. We have not used any of the graphital oils, neither have we used special oils that are supposed to increase film strength.

No mention has been made of oil reclamation in Mr. Faulkner's paper. In all of our larger garages we have oil reclamation and know that it has saved us considerable money. We feel that the reclaimed oil is equal to new oil.

Operating Phase Neglected by Manufacturers and Refiners

—**Clinton Brettell**
R. H. Macy & Co., Inc.

IT has been my impression, gained at several S.A.E. meetings on lubrication, that the manufacturers, or at least their delegated representatives in attendance at meetings, considered that there was no operating problem concerned with lubricants, and that production and distribution were all that mattered.

It almost seemed that they resented the attempts of operators to pry into the "chamber of horrors" built around automotive lubrication.

I, therefore, wish to reiterate what I have said before at some of these meetings, namely:

That automotive lubrication is as much an operating as a manufacturing problem.

That, if lubricants were not used by operators, there would be no need to either produce or distribute such products.

That, therefore, the operating phase of automotive lubrication is highly important and should receive more frank and cooperative treatment by manufacturers, both lubricant and vehicle.

Standards for New and Used Oil.—Specifically, I agree with Mr. Faulkner, that definite standards (limiting values) based on research (laboratory and field, coordinated) should be set up to show:

(1) The essential physical properties of an oil for engine lubrication (with the non-essentials eliminated).

(2) Essential physical properties to determine when an oil is no longer fit for further use in the crankcase (without reconditioning outside of crankcase).

These standards should properly cover typical classes of operating conditions, such as long runs, high speeds, city multi-stop, cold weather, hot weather, trucks, passenger cars, bearing metals, filters, and so on.

Viscosity Index and Oxidation.—I agree with Mr. Faulkner that viscosity index (a rather hazy term, which even many "oil men" seem unable to define in simple language) and oxidation tests are probably among the most important characteristics, although the flash point (indicative of volatility and hence consumption) and fire point (in some measure indicating ability to "burn clean") might still be considered as meriting some consideration.

Just how important acid reaction is, and whether the kind of acid is worthy of note, is now, and always has been, a very doubtful question in my mind. It certainly did not seem to be a prime factor with the old bearing metals, at least not in the limits with which I have had experience. Also, does it or does it not promote sludging, and does it aid in "penetrability" of the lubricant?

You can get many opinions on this subject, but apparently nothing supported by research.

Sludge.—On the question of an increased sludging tendency with the lighter viscosity oils, which Mr. Faulkner reports, I would say that our experience does not seem to check on this point. In fact, we seem to find less sludging than formerly.

How much of this decreased sludging can be attributed to each of the following factors is difficult to evaluate, although we noticed a distinct improvement with the change to re-refined oil:

(1) Improvement in inherent characteristics of the oil (re-refined oil used on about one-half of the fleet).

(2) Better filters and more careful servicing of them at periods of 2000-4000 miles, based on a blotter test of oil.

(3) Better engine design.

(4) Improved operating temperatures.

Drain Period.—We ran oil in typical cars of various makes and under a variety of operating conditions, and had the crankcase oil (sample drawn from line from circulating system) checked in our laboratories at various mileages.

Based on the limits for various contaminants, such as we were able to secure from oil refiners, we rated these oils as "fit" or "unfit" for further use.

We found that up to 5000-6000 miles on an engine in good condition and reasonably long daily mileage, the imposed limits were not exceeded. Above that point we would find an occasional case where they were exceeded. We, therefore, decided that for engines in poor condition this limit should be about 2000 miles.

On short runs and for good engines, 2000-3000 miles was the limit and, where the engine was in poor condition, it was about 1000 miles.

We, therefore, set these limits as guides for our garage supervisors to work to, modified by their judgment in special cases where these limits seemed too high.

We found many cases where oil was rated as fit for use after 10,000 to 12,000 miles, and we are still very far from satisfied that the standards on contaminants are really essential or properly established. In fact, we wonder whether the oil could not safely be used indefinitely.

Some fleets are working on exactly that basis and claim to find "no measurable deterioration in engine condition" from such practice. We are trying, on a very limited scale, to research this phase of the problem ourselves. Possibly we could not, in fairness, expect the oil companies to conduct tests that would tend to reduce their sale of lubricants, although competition will probably force this issue before long.

Grade of Oil.—We do not seem to check with Mr. Faulkner that "fleets are going back to higher priced oils again." Many of them in this locality seem to be working in just the opposite direction, in some cases using the "new process," extremely low-cost oils (reputedly de-sludged). I would be very much interested to learn the reaction of this meeting on that subject.

Chassis Lubricants.—We heartily agree with Mr. Faulkner that there is a crying need for research, field and laboratory, leading to simplified requirement standards that can be readily applied to the average fleet, and necessitating a minimum of products.

Although, admittedly, such a procedure will result in a compromise that will not meet any specific condition 100 per cent, it is far preferable to one that would select the "ideal lubricant" for each particular purpose, and then require a "lubrication engineer" at each garage to see that the correct lubricants were used in each and every instance. This example might sound far-fetched, but it is a practical condition that has to be met in every fleet of appreciable size and decentralization.

E. P. Lubricants.—We question the need for this product in many fleets. The company (one of the largest in the country) which has the contract for our lubricants made a survey of our fleet (410 vehicles, 1909 to 1935 models, mostly 1930-1935), and in no instance did they recommend an E. P. lubricant.

After operating on their recommended basis for one year, we encountered difficulty with only one model, a tandem-axle, dual-worm drive (24-hr. service, small reservoir capacity), and the oil company has now recommended E. P. lubricants for this vehicle. We were also advised that ours was not an unusual operation but, rather, typical of many fleets.

Although we have encountered the rapid transmission-gear wear (not scoring) reported by Mr. Faulkner (the most extreme case being 2000 miles with one cheap, light truck in its original design), we attributed it to extremely small wearing surfaces coupled with improper design, causing excessive clashing. We question very seriously whether any E. P. lubricant would have materially improved this service.

It is our understanding that E. P. lubricants have certain distinct disadvantages, and we would, therefore, be inclined, particularly when supported by the oil refiner, to refrain from using them except where it is clearly dictated.

Water Pump Lubricants.—We concur with Mr. Faulkner's statement that this matter is becoming very involved due to the several designs for pump lubrication now being furnished on various makes of trucks. We would strongly urge that one standard system of pump lubrication and lubricant be worked out and put into immediate operation.

Fan Lubrication.—The same comment holds here, although the reservoir-oil type (which is generally satisfactory) now seems to be destined for early standardization.

Transmission and Rear-End Lubrication.—We find that two lubricants care for all but five of our gasoline cars and two for our electrics (housed at a separate garage), and this method is a reasonably simple routine. We make a change, of course, between summer and winter grades. We use this same product for steering-gear and general chassis lubrication.

Universals and Wheel Bearings.—We use a high-pressure fiber grease for this purpose, applying it by hand at most of our garages. Where we have a gun that will handle it, we also use this same product for general chassis and steering-gear lubrication and, consequently, obtain a much longer interval between necessary lubrication jobs.

Compromise Possible on Number of Lubricants

—Henry L. Debbink
The Milwaukee Electric Railway and Light Co.

MR. FAULKNER definitely has the operator's viewpoint in approaching the problem of lubrication and outlines this viewpoint well. However, I think that he is both a little too hard and, in some respects, too easy on the refiner and manufacturer.

He is a little too hard on them in making so much objection to the great variety of manufacturers' specifications for different makes. He knows that, if we are confronted with specifications which require six different viscosities in one garage, we will pick the best compromise if we are limited for practical reasons to two or three viscosities. Moreover, this compromise will represent just as satisfactory fleet lubrication as if the attempt were made to follow the manufacturer's recommendations with detailed exactness.

He is too easy on the refiner and manufacturers in accepting so many of their recommendations for lighter engine oils and E. P. gear lubricants. It is true, as Mr. Faulkner points out, that connecting-rod bearings have given more trouble in recent years. There have been a number of reasons for this trouble, but could not the use of lighter oils be responsible for some of the fatigue of bearing metal in the top half of the connecting rod in heavy-duty service?

In regard to E. P. lubricants, I think the conflicting results still indicate that the fleet owner is justified in taking the position that he should use them only in special cases where tooth pressures are so high as to make their use mandatory and that, in general, a high-quality conventional lubricant is best. The quality of this lubricant will depend on the petroleum stock in it. For this purpose I like a high-quality well-refined steam-cylinder stock of at least 220 sec. viscosity at 210 deg. fahr. for buses or heavy-duty equipment.

In regard to the question of when an oil is unfit for further use, Mr. Faulkner's suggestion that crankcase oils be discarded if they show more than 1 per cent of material insoluble in petroleum ether is an interesting specification, but it may not be "tight" enough. I have checked 149 crankcase oil analyses made last year on samples from our buses. These samples had been run an average of 3000 miles, though some on special test had been run as much as 14,000 miles. I find that only two of these 149 samples showed more than 1 per cent insoluble in petroleum ether.

Probably this question should be studied further. Also, there should be agreement on whether volume or weight measurements of petroleum ether insolubles are to be made as some concerns have made measurements in both ways.

Chemical Reactions May Explain Lubrication "Mystery"

—**Alfred F. Coleman**
Consulting Engineer, New York City

IT is gratifying to note that Mr. Faulkner emphasizes the Pacific Coast and Gulf State climatic conditions which are such as to justify the year around use of summer grades, and also such as to justify certain refiners for asking for leeway under the S.A.E. lubricants set-up so that the winter oils they are forced to supply by virtue of manufacturers' recommendations can be made without prejudice from their native crudes. Why force the supply of high V.I. oils such as the restrictive features of the 20-W and 10-W program demand in localities where such oils are not required by service conditions?

In his reference to the correlation between starter, battery, and characteristics of the oil recommended by the car manufacturer, Mr. Faulkner brings to the surface a relationship that will bear much study. Here we are dealing with measurable units of force and resistance. Why should there be the present guesswork?

In his frequent references to relative sludging tendencies of different grades of oil, to the limitations of conventional oil specifications, and to the relationship of crankcase oil analysis to the service expectancy of the oil, Mr. Faulkner, in my opinion, opens up a jackpot and affords the opportunity to point out a technical truth which often escapes both factory engineer and the fleet operator. This truth is that the behavior of oils in service depends on basic chemical reactions which vary widely with different oils in identical service and equally widely with the same oil in different classes of service.

In this statement lies the reason for much of the "mystery" surrounding lubricants and their performance. Their fundamental chemical composition is not known except in very general terms, and the basic chemical reactions to which their component parts are subjected under varying temperatures, oxidizing conditions, and so on, are not understood. The mystery, of which the operator complains and which many are trying to dispel, will not lose its ethereal form until petroleum research workers solve these two allied problems.

In this connection, I feel that it is timely to point out that many engineering salesmen representing real up-and-going refiners are well equipped to supply answers to most of the operators' pertinent and pressing questions. Often the fleet operator cuts himself off from such technical assistance by deluding himself with the idea that he is being "high-pressured," or by milking the technician dry and then buying his lubricants on a price or a specification basis either of which defeats the very ends he seeks. It costs money for a refiner to educate an engineering staff up to a point where they can render real service to the fleet operator, and it just is not economical to supply that type of service to price and specification buyers.

Mr. Faulkner's point on the cost of maintenance as against the cost of lubricants, coming from an operator instead of an oil man, will doubtless do much to emphasize this point. In fact, the general tone of his whole paper should act in this direction.

On filters, I wonder how many operating men have studied the relative rates of flow of oil through filters taken at periodic intervals. The results are surprising and certainly indicative of the need for larger and better filters if we are to reach the results we hope to reach through their use.

It is most gratifying to have Mr. Faulkner emphasize the fact that manufacturers in general this year are recommending all grades of oil from 10-W plus 10 per cent of kerosene all the way up to S.A.E. 50 under graduated, but overlapping temperature ranges that require the expert services of the U. S. Weather Bureau to interpret and apply.

This sort of thing and the discrepancies between what is now current and what was recommended last year and the year before do much to discount in the minds of operators all the emphasis the manufacturer places on lubrication instructions.

Erratum

In Fig. 15 on page 79 of the Transactions Section of the February issue of the S.A.E. JOURNAL, in the article by Neil MacCoull and G. T. Stanton, 300 cycles in the lower shaded portion of the drawing should have been 3000 cycles.

Engine Nacelles and Propellers and Airplane Performance

By Donald H. Wood

Aeronautical Engineer, National Advisory Committee for Aeronautics

THE extensive experimental study of engine-nacelle location and cowling made by the N.A.C.A. is reviewed. The factors contributing to the efficiency of the engine-nacelle and propeller group are discussed. Several examples are given showing how the results of the experiments are applied, and how the use of improved cowlings and better nacelle locations increases the high-speed performance of airplanes. It is shown that about 25 per cent of the improvement of present-day airplanes over those of seven or eight years ago can be attributed to the increased aerodynamic efficiency of the engine-propeller group alone.

The effect of engine size on the nacelle drag is discussed, and charts are given from which values may be used in calculating the nacelle drag in preliminary performance estimates. Whether other types of engine, in-line air-cooled and liquid-cooled, can compete with the air-cooled radial is shown to depend on the provision of cooling arrangements of low drag. The question of radiator resistance is of great importance and requires extensive study; but there is reason to believe that radiator drags can be reduced to less than half of present values, which will increase the field of use of the liquid-cooled engine.

The use of engines mounted entirely within the wing, with the propeller driven through an extension shaft, has been suggested. A few experiments are described and it is shown that considerable improvement is to be expected if the engine can be cooled by methods that do not increase the drag. Further experiments with these arrangements seem to be worthwhile and offer the greatest possibility of improvement over existing types.

IN the development of the airplane from its earliest construction, when it was a question of whether or not a particular design would fly, to the time when some measure of performance was demanded, the requirements were met by the installation of engines of greater horsepower. The improvement of the airplane and the increase in its load-carrying ability and its aerodynamic efficiency have been connected throughout the years with the development of greater power, higher efficiency, and increased reliability of the airplane engine.

Demand for Higher Efficiency.—The early airplane designs were mainly for military use; economy of operation was of secondary importance compared with performance. Such commercial designs as appeared were usually operated for exhibitions or for sport. The cost of operation was again not of first importance. When the airplane had begun to establish itself as a competitor of other modes of transportation, however, it became evident that every advantage must be taken of methods that would decrease its operating cost; all the more so because more passengers had to be carried in one unit and their greater comfort demanded roomier fuselages; furthermore, the resulting larger airplane had to be moved through the air at increasingly higher speeds. All of these and many other requirements demanded more power or improved efficiency. Actually, the modern multi-motored airplane had some of each of these two properties, but high power now has to justify itself in dollars and cents. It is clear, however, that the improvements which have shown themselves in the past few years would not have been made without the stimulus of commercial transport aviation, and would not have far exceeded the dreams of ten years ago when this stimulus was only well begun. The efficiency of practically all elements of the airplane has been increased, and each improvement has contributed in some measure to present-day performance and economy.

Development of Cowlings.—Among the numerous elements, and one of the most essential, is the engine. The radial air-cooled engine, which has attained such importance in air transport, offers a very high resistance to its passage through the air. One of the first important improvements was the development of cowlings, notably the N.A.C.A. cowling, which so directed the air flow around the engine

[This paper was presented at the Annual Meeting of the Society, Detroit, Mich., Jan. 16, 1935.]

Fig. 2—Airplane with Old-Type Engine Nacelles

that the resistance was greatly reduced. This feature was readily applied to the engine located in the nose of the fuselage but, when it was applied to the outboard engines of multi-engined airplanes, the increase in performance either was not startling or, more often, no improvement was noticed. The reasons for such results are now quite clear, but six years ago they were obscure and many experiments were required to clarify them.

Further Research Required.—The first indication of difficulty came when indifferent results were obtained in tests of a Fokker Tri-motor airplane on which N.A.C.A. cowlings were installed in the spring of 1929 following their successful application to a single-engine training plane the previous year. The experiments reported by Jacobs[1] were, therefore, made as a start in clearing up the difficulty. In May, 1929, at the annual Aircraft Engineering Research Conference, Col. V. E. Clark suggested that the National Advisory Committee for Aeronautics investigate the effect of placing the engine nacelle and propeller in various positions forward of a wing. This suggestion was predicated on the idea that some unfavorable interference with the air flow was causing the poor results as had been shown by the above-mentioned experiments[1], and that some more favorable position or combination might be found. Furthermore, the propeller, which was later shown to have important effects, was to be included. Previous to this time, the location of engine nacelles had been a matter of individual design, the most common arrangement being a nacelle suspended below the wing with the propeller somewhat forward of the leading edge. Undoubtedly this arrangement had many practical features to recommend it and, under the speed and power conditions of the time, it did not appreciably reduce the performance.

Scope of the Investigation.—The suggestion cited resulted in a program of tests which, with many extensions in detail, has been going on ever since. The results have been given in several N.A.C.A. reports, each covering a somewhat arbitrary division of the subject. Monoplane and biplane wings; radial, in-line, air-cooled, and liquid-cooled engines; numerous cowling forms; and tractor, pusher, and tandem propellers have been tested in various combinations in the experiments to date. It is the purpose of this paper to discuss some of these experiments, the factors involved, and their relation to airplane performance. Some ideas of possible use in future designs will also be developed from the analysis.

[1] See N.A.C.A. Technical Note No. 320; "The Drag and Interference of a Nacelle in the Presence of a Wing", by Eastman N. Jacobs.

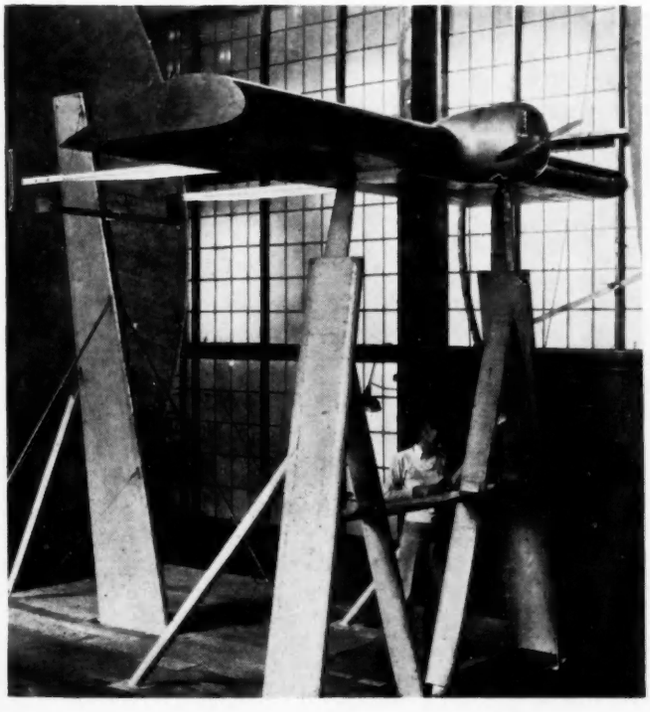

Fig. 1—Typical Nacelle Set-Up in N.A.C.A. Tests

Mutual Influence or Interference

At the outset it will be instructive to recall certain principles which, although elementary, are often lost sight of in dealing with problems. Whenever two or more objects are placed near one another and caused to move together through a fluid, the force acting on the combination of objects is, in general, different from the sum of the forces that would act on each object if moved through the fluid alone. In effect, the several objects become a single new object, or all the objects may be said to have mutual influences on each other. A common tendency is to assume that the mutual influence is lacking in cases where it is unknown, and this wrong notion is often carried to extremes even in cases where common sense dictates otherwise; it must be admitted, however, that this assumption is made more from necessity than from choice. It is important to recognize the existence of the mutual influences as well as the fact that the lack of them is the exception and not the rule. In many cases the mutual influences appear to be lacking when, in fact, the influences merely neutralize each other.

These old and well-known facts are stressed because so many people who are very solicitous of the effects among some parts of the airplane, calmly ignore equally large and important ones among other parts. For example, it is idle to expect the same result from a nacelle located above a wing by itself as from a wing and nacelle mounted above a large boat hull. If a poor result is obtained, the nacelle is blamed, although the hull is by far the largest object in the picture and may be the most serious offender. It is to be understood, therefore, that the discussion of this paper is

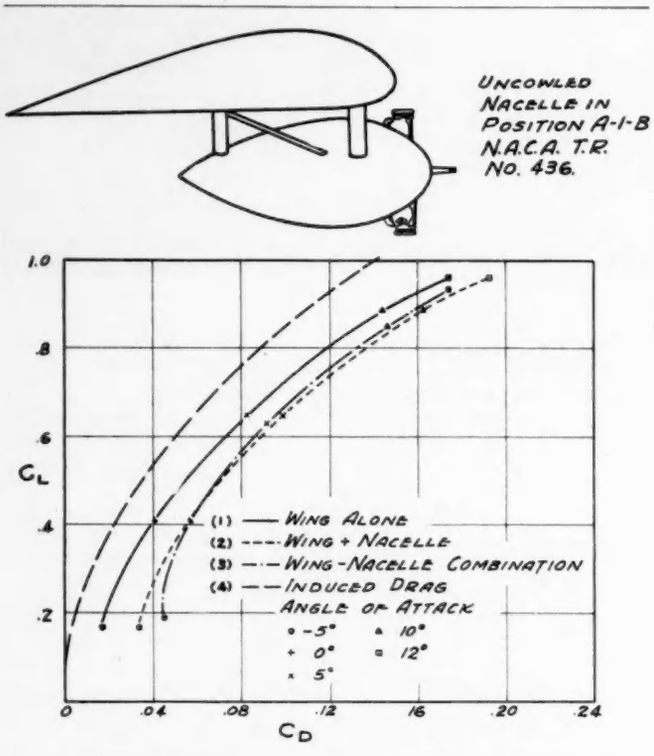

Fig. 3—Nacelle Drag and Interference; Wing with Aspect-Ratio 3; Uncowled Nacelle

confined to the mutual influence of wings, engine nacelles, and propellers; and that other important parts of the airplane are excluded, not because they are unimportant or are unrecognized but because they are, for the moment, secondary to this necessarily limited analysis.

Interference.—The mutual influence between bodies is most commonly called interference. It is customary to describe the total forces and moments in terms of their components, and so there may be interference lift as well as interference drag, and there may also be interference moments. If the total drag of a combination is greater than the sum of the drags of each isolated body, the interference is called unfavorable or positive. If the total drag is less than the sum, then the interference is favorable or negative.

It is apparent that interference is likely to arise at many points of an airplane. Research[2-4] on the general subject has been quite extensive. The wing and fuselage, the fuselage and tail, and the landing gear have been studied in great detail. Tests have shown the sources of positive interference and suggested means for its reduction and, in some cases, its elimination, and also for the production of negative interference with resulting improvement in performance. The positive interferences are often surprisingly large, and their reduction is equal in importance to that of reduction of the resistance of the several bodies making up the complete airplane. The wing-nacelle problem enters both fields. Cowling and fairing reduce the resistance, and proper location reduces the interference.

Cause of Interference.—The interference may be considered to arise from turbulence resulting from divergent flows caused by the particular disposition of the bodies; that is, from changes in the pressure and velocity distribution causing changes in the local pressure and the skin-friction drags. Often two bodies are touching or one may run into the other so that there is a reduction of the total surface exposed to the air-flow. Yet the interference may be unfavorable because increased turbulence or pressure and friction forces may arise. In fact, this case is one of the most important because, in all practical constructions, the various bodies must be attached to each other, at least by struts or wires, which in themselves may introduce interference.

The Engine Nacelle.—Some of the early airplanes were equipped with engines suspended by means of struts and wires and without any fairing or housing whatsoever. The introduction at the rear of the engine of a fairing section, however small, makes the combination an engine nacelle. It has long been recognized that dead areas at the rear of bodies must be eliminated so far as is possible and the most important portion of the so-called streamline body is the after-portion. Reasonable bluntness at the nose is not serious. Before the introduction of cowlings, the engine cylinders of the radial engine constituted a series of bumps or excrescences that introduced turbulence and, consequently, high drag. The expansion tank, the radiators, and the exhaust pipes have similar effects on the liquid-cooled engine nacelle. The elimination of the bad effects of such apparently minor appendages must also receive attention.

The radial air-cooled engine has attained such prominence and is in such universal use in air transport that many of the tests were made with this type of engine. Indeed, it will be shown later that the air-cooled engine does not suffer aerodynamically when compared with the liquid-cooled engine with its attendant radiators.

The Propulsive Unit.—The engine, its housing or nacelle, and the propeller constitute what will be termed a "propulsive unit." These three elements are complementary. If one could be eliminated, all could be. The propeller converts the torque produced in the engine into thrust, which overcomes the drag of the airplane. Engines may assume many forms and positions, and the propeller may be driven through a shaft, long or short. All powerplants must be cooled by contact of some part of the system with the air. In a proper assessment of the over-all efficiency of the propulsive unit, the drag of the nacelle, the drag due to the cooling apparatus and the efficiency of the propeller must all be included. Any comparison that does not take all these factors into account is misleading and of no value unless qualified by a suitable explanation. Exaggerated claims are made for many devices in all arts, substantiated by an incomplete presentation of the facts, and aviation is not immune from them. It will be necessary to be a party to this subterfuge later on, but it is hoped that "suitable explanation" will have been made.

Problem and Method

The drag of the nacelle and cooling apparatus arises from the skin friction and from the pressures created on the surfaces by the air-flow. The proximity of the nacelle and the wing affects the velocities and directions of the air-flow about each of these parts with resulting changes in the forces. The slipstream from the propeller introduces additional effects, and the propeller is in turn influenced by its location in the field of flow. The problem is to find out something about the nature and magnitude of these effects.

Experimental Methods.—One method of determining the

[2] See British A. R. C. Reports and Memoranda No. 1480; "Some Aspects of Mutual Interference between Parts of Aircraft", by E. Ower.
[3] See *The Journal of the Royal Aeronautical Society*, July, 1932, pp. 531-577; "Interference", by E. Ower.
[4] See *Aircraft Engineering*, April, 1934, pp. 93-96; "A Note on Interference", by E. Ower.

magnitude of the forces is to measure them on each body when alone and then on each body when in the presence of the others. This method yields results of great value, but is often attended with experimental difficulties. From the practical viewpoint a somewhat simpler method giving useful results almost directly may be used. Applied to the wing-nacelle problem the procedure is about as follows:

The forces on the wing alone are first measured. Then the nacelle is installed and the forces on the combination are measured. The difference in the two sets of forces shows the effect of the nacelle on the wing and, if the forces on the nacelle alone are also determined, another difference gives the effect of the wing on the nacelle. Finally the propeller is attached, and its characteristics are determined. The isolated propeller is not usually tested, because such a test is not required and would be, indeed, a difficult problem by itself.

Precautions.—If one is interested only in the overall performance, a sufficiently comprehensive test with the propeller operating is sufficient. Such a test gives the total efficiency but sheds practically no light on the contributing factors. It also has the disadvantage that the forces due to the nacelle are such a small proportion of the total that the effect of small alterations is not detected. In fact, when the model is limited to only a part of the complete wing and the nacelle, attention must be paid to accuracy. The propeller presents similar problems in that the higher speeds have to be simulated by reducing the revolution speed of the propeller because of the limited air speed of most wind tunnels. The forces due to the propeller then become small, and the resulting accuracy is relatively poor. The inventor's dream of efficiencies from 125 to 150 per cent is easily attained with practically no power input unless precautions are taken.

General Features of N.A.C.A. Tests

The research conducted by the N.A.C.A. has followed the general lines indicated previously. In order to make the results of the greatest comparative value a large number of the nacelle arrangements have been tested on a single wing. Enough tests with other wings have been made, however, to show that the effect of a change in wing section is not of predominant importance. The wing used was of reduced span in order to make the forces due to the nacelle and propeller as large a proportion of the total forces as was practicable. A typical combination in test position in the 20-ft. wind tunnel is shown in Fig. 1. Subsequent tests have shown that substantially all the effects were present on the reduced span.

Scope.—The nearly 160 combinations tested to date have included 21 N.A.C.A. cowled-nacelle locations with tractor propeller ahead of the wing, 6 positions with several types of cowling, 11 positions with tandem propellers, 17 positions with pusher propellers, 12 positions on a biplane wing, and several miscellaneous arrangements including in-line engine nacelles and extension-shaft schemes.

The test data have been reduced to manageable proportions before being published as reports.[5-12] Some parts of the series have not been completed. These results are as yet unpublished, and extensions to the program are still being made.

Although the data are presented in a form that can be used in design calculations, a discussion of some of the facts will save labor as well as be of assistance in clearing up questions that designers have not been able to bring to the attention of the investigators. Some questions that have arisen will be considered on the assumption that they may be of interest to others.

The Designer's Problem

Of first interest, of course, is the arrangement of nacelle, cowling, and propeller that gives the highest net efficiency.

[5] See N.A.C.A. Report No. 415; "Tests of Nacelle-Propeller Combinations in Various Positions with Reference to Wings, Part I, Thick Wing—N.A.C.A. Cowled Nacelle—Tractor Propeller", by Donald H. Wood.

[6] See N.A.C.A. Report No. 436; "Tests of Nacelle-Propeller Combinations in Various Positions with Reference to Wings, Part II, Thick Wing—Various Radial-Engine Cowlings—Tractor Propeller", by Donald H. Wood.

[7] See N.A.C.A. Technical Note No. 432; "Drag Tests of 4/9-Scale Model Engine Nacelles with Various Cowlings", by Ray Windler.

[8] See N.A.C.A. Report No. 462; "Tests of Nacelle-Propeller Combinations in Various Positions with Reference to Wings, Part III, Clark Y Wing—Various Radial Engine Cowlings—Tractor Propeller", by Donald H. Wood.

[9] See N.A.C.A. Technical Note No. 420; "The Effect of Propellers and Nacelles on the Landing Speeds of Tractor Monoplanes", by Ray Windler.

[10] See N.A.C.A. Report No. 505; "Tests of Nacelle-Propeller Combinations in Various Positions with Reference to Wings, Part IV, Thick Wing—Various Radial-Engine Cowlings—Tandem Propellers", by James G. McHugh.

[11] See N.A.C.A. Report No. 506; "Tests of Nacelle-Propeller Combinations in Various Positions with Reference to Wings, Part V, Clark Y Biplane Cellule—N.A.C.A. Cowled Nacelle—Tractor Propeller", by E. F. Valentine.

[12] See N.A.C.A. Report No. 507; "Tests of Nacelle-Propeller Combinations in Various Positions with Reference to Wings, Part VI, Wings and Nacelles with Pusher Propeller", by Donald H. Wood and Carlton Bioletti.

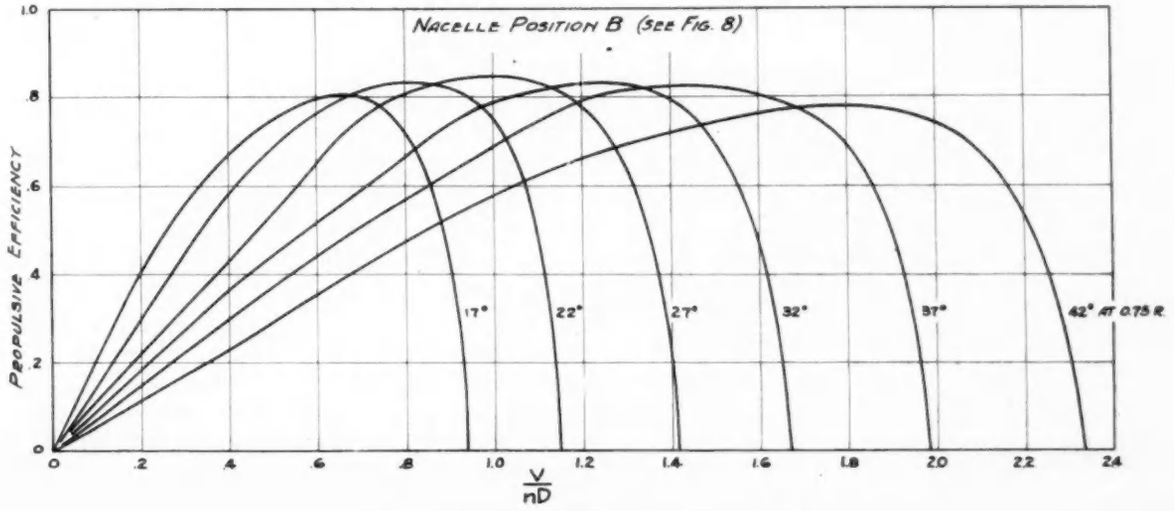

Fig. 4—Propeller Efficiency Against V/nD

It would be a temptation to stop here and consider the problem solved. Different classes of service, however, demand different airplanes, and considerations other than efficiency often dictate the main features of the design; therefore, the nacelle cannot always be located in its best possible location. Furthermore, there is no guarantee that the best arrangement has been found. It is necessary, therefore, to have fairly complete information on many nacelle locations and cowlings in order to satisfy the demands of design. It is not sufficient to know that one location is inferior to another—we must know also how inferior it is. It will be admitted, however, that the less efficient locations are not so much in demand, and discussion of the superior ones will point the way to the solution of all, so that no difficulty should present itself in handling cases not specifically discussed if the original data are referred to.

Improved Efficiency.—The improvement in performance that has resulted from the study is best indicated by comparing the performance of a hypothetical airplane equipped with uncowled radial engines suspended below the wing with that of the same airplane with cowled nacelles ahead of the leading edge. The latter location was found superior in the first part of the series of tests[5] and, in fact, has not been improved upon to date for radial-engine installations.

Old Type Airplane, Examples I to III

In order to show how the performance estimates are obtained and, at the same time, to explain certain features of the method more clearly, an example will be worked out in detail. A two-engine airplane similar to the one shown in Fig. 2 will be considered to avoid the conditions arising when the engine is in the nose of the fuselage, a location foreign to this discussion. The problem of the three-engine airplane can, of course, be treated by methods similar to the one outlined here.

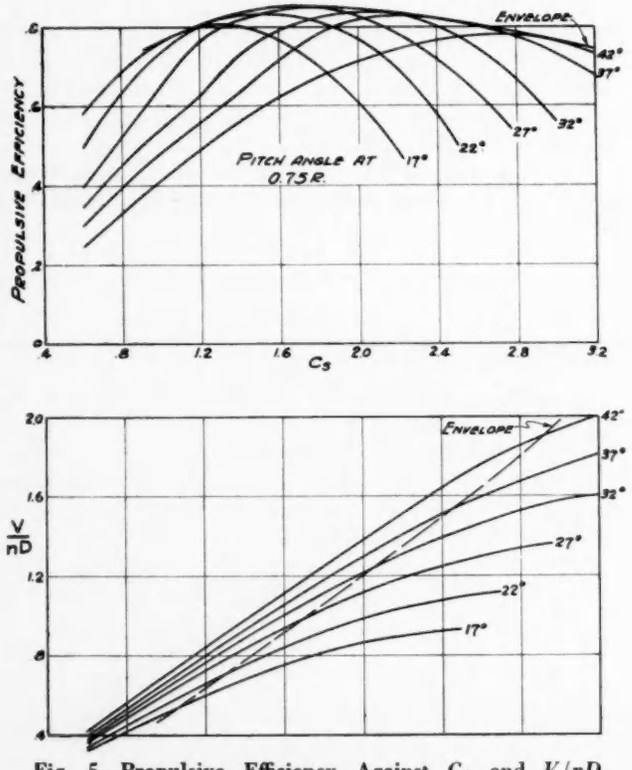

Fig. 5—Propulsive Efficiency Against C_s and V/nD Against C_s

The airplane has the following assumed characteristics:

Total weight	12,125 lb.
Wing area	882.6 sq. ft.
Wing span	75.5 ft.
Engine power	2 engines, each developing 555 hp. at 1950 r.p.m. at sea level, 53¾ in. in diameter
Profile-drag coefficient (without nacelles)	0.0344

From these values there results:

Wing loading	13.7 lb. per sq. ft.
Power loading	10.9 lb. per hp.

The profile-drag coefficient is here assumed, but it may be obtained by summing up the drag of the parts of the airplane (with judicious estimates of the numerous interferences) or, preferably, from a wind-tunnel test of a model *without* engine nacelles. Note that the drag of the nacelles is not included, nor are the additional effects of the propeller. These values come in at a later stage.

Old-Style Nacelle.—From test results[6-7] the curves of Fig. 3 have been plotted. The standard polar curves of lift coefficient against drag coefficient are given (1) for the wing alone,[6] (2) for the sum of the wing-alone drag and nacelle-alone drag,[7] and (3) for the wing-nacelle combination illustrated. This combination represents the old uncowled engine-nacelle location. Earlier in this paper the interference question was discussed. The difference in drag between curve (3) and curve (2) is the interference drag. It will be noted from examination of the points at a given angle of attack that there are both interference lift and interference drag. The angle of attack is largely a necessary evil in aeronautics, and serves no very useful purpose except as a reference point. If the differences in drag are taken at the same values of the lift coefficient, the interference lift may be dismissed. This procedure falls down at high angles of attack, where the maximum lift of the combination may be different from that of the wing alone, but is very useful throughout the remainder of the range.

Although the difference in the values plotted in curve (3) minus those of curve (2) represents the interference, the difference in the values of curve (3) minus curve (1) is of greater interest because it represents the drag added by the nacelle, namely, the "effective" nacelle drag, which is required in determining the total airplane drag.

The curves are not parallel in the range of C_L from 0.2 to 0.4 covering the high-speed and cruising range and the use of a single value for the effective drag will be in error. At lift coefficients from 0.5 to 0.6, the added nacelle drag is a relatively small proportion of the total and will be a still smaller proportion in an actual airplane because the wing area is generally larger in proportion than in the reduced-span model tested. The added nacelle drag will, therefore, have a relatively small effect on performance in climb, which is found to be generally true of all drag-reducing methods. The effect on climb is quite small, owing to the predominant influence of the induced drag, as can be seen from the induced-drag polar curve (4) also given in Fig. 3. The induced drag is somewhat exaggerated here, however, because the curve (4) is plotted for aspect ratio 3 and includes the induced drag due to tunnel-wall interference, which is positive in the open-throat tunnel used.

High Speed.—The performance of an airplane is usually

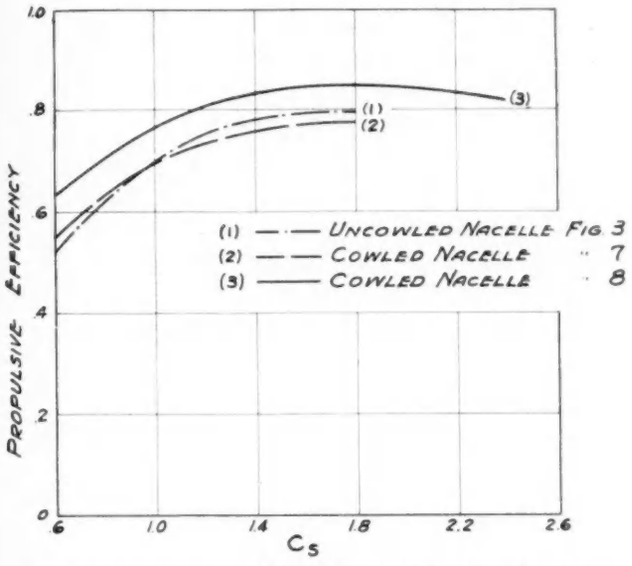

Fig. 6—Propulsive Efficiency against C_S; Envelope for Three Nacelle Locations

expressed in terms of the high speed or the cruising speed, maximum rate of climb, and landing speed. To determine the high speed the following equations may be written:

Power required = Power available

Drag × velocity = Motor power × propulsive efficiency

or

$$\left(C_{D_P} + C_{D_N} + C_{D_i}\right) S \times \frac{\rho V^3}{2} = P \times \eta \qquad (1)$$

where the subscripts P, N, and i mean profile, nacelle, and induced, respectively, and the other symbols are standard. The value of C_{D_N} taken from Fig. 3 in the manner already described, at $C_L = 0.325$, is:

$$\text{Model } C_{D_N} = C_{D_C} - C_{D_W} = 0.0500 - 0.0300 = 0.0200$$

where the subscripts C and W refer to combination and wing respectively. This value is based on the model wing area and is to model scale and must be converted to the full-size engine cross-section area and to full-size wing area thus:

$$C_{D_N} = \text{Model } C_{D_N} \times \left(\frac{\text{model wing area}}{\text{full size wing area}}\right) \times \left(\frac{\text{airplane engine diameter}}{\text{model engine diameter}}\right)^2$$

The model wing area is 75 sq. ft.; the model engine diameter is 20 in. The full-size engine diameter is 53.75 in., and the full-size wing area is 882.6 sq. ft., as previously given.

Then

$$C_{D_N} = 0.0200 \times \frac{75}{882.6} \times \left(\frac{53.75}{20}\right)^2 = 0.0123$$

The induced-drag coefficient now has to be calculated. This calculation will depend on the wing span load distribution. For simplicity, since it is a minor feature of this problem (only comparisons are being made), elliptical distribution will be assumed.

Then induced drag $= \dfrac{\text{lift}^2}{\pi q b^2}$

$$C_{D_i} = \frac{C_L^2 S}{\pi b^2}$$

where b is the wing span (in general, "effective" span), S the wing area, and q the dynamic pressure. The lift coefficient is not known, so it is more convenient to rewrite equation (1) as

$$\left(C_{D_P} + C_{D_N}\right) S \frac{\rho V^2}{2} + \frac{\text{lift}^2}{\pi q b^2} V = P \times \eta \qquad (1a)$$

or

$$\left(C_{D_P} + C_{D_N}\right) S \frac{\rho V^3}{2} + \frac{2 \times (\text{lift})^2}{\pi \rho V b^2} = P \times \eta$$

The equation is to be solved for V, and the only remaining quantity to be determined is the propulsive efficiency η.

The Propeller Question

Efficiencies.—A digression will be made at this point to clear up some points related to propellers. If a propeller is operated and the tension in the shaft T is measured along with the power input, an efficiency $\eta_P = \dfrac{T \times V}{P}$, called the "propeller" efficiency, is obtained. This efficiency has little practical use because the slipstream acts on all bodies behind and changes their resistance and the propeller has to overcome this added resistance.

If the slipstream drag is deducted from the thrust, an effective thrust T_e is obtained and the "propulsive" efficiency $\eta = \dfrac{T_e V}{P}$ is obtained. This efficiency is the most useful. The slipstream drag ΔD is not easily obtained directly, but T_e is simply obtained by adding to the force on the whole propeller-body system with propeller operating R the force obtained with the propeller removed D. $T_e = T - \Delta D = R + D$. Note that ΔD is not the whole drag but only the drag due to the action of the propeller. The propulsive efficiency affords a convenient method of taking care of troublesome slipstream corrections. It is in practically universal use in this country and has been determined for a great many propeller-body combinations both full-scale[13] and model.[5-12, 14]

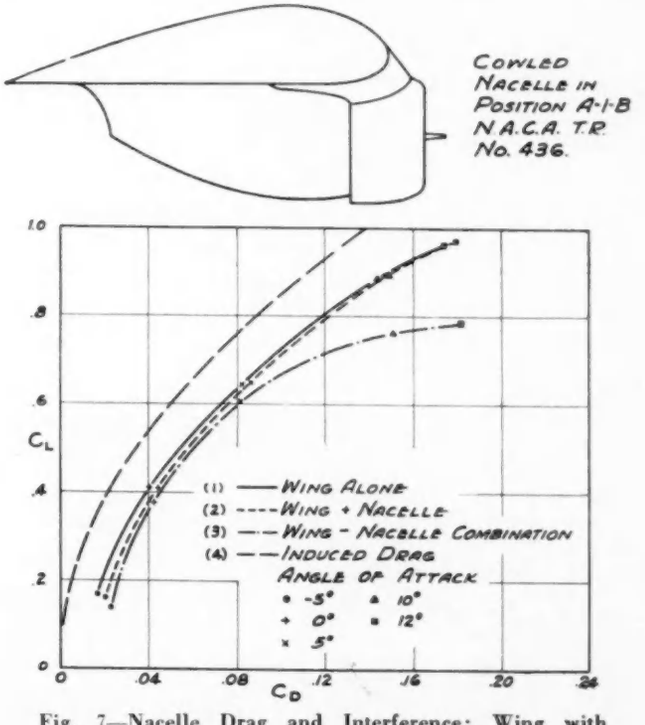

Fig. 7—Nacelle Drag and Interference; Wing with Aspect Ratio 3; Nacelle Faired into Wing

[13] See N.A.C.A. Report No. 350; "Working Charts for the Selection of Aluminum-Alloy Propellers of a Standard Form to Operate with Various Aircraft Engines and Bodies", by Fred E. Weick.

[14] See N.A.C.A. Technical Note of Nov. 15, 1935; "Tests of a Wing-Nacelle-Propeller Combination at Several Pitch Settings up to 42 deg.", by Ray Windler.

If the resultant force on the propeller-body combination with propeller operating R is used, a "net" efficiency is obtained: $\eta_n = \frac{R \times V}{P}$. This efficiency is very useful in certain comparisons, but must be used with caution because it will assume many values depending on how much of the airplane is included in the test. For example, if a complete airplane is tested with propellers operating the net efficiency is zero at the high speed of the airplane, for the forward force just equals the drag. If only the effective nacelle drag D_N is deducted from the effective thrust, a net efficiency $\eta_n = \frac{(T_e - D_N)}{P} V$, is obtained which indicates the part of the engine power that remains after deducting the propeller losses and the nacelle drag. This kind of net efficiency has been used in comparing nacelle-propeller arrangements and will be discussed later.

The efficiency of the propeller varies within wide limits depending on the pitch, forward speed, and revolution speed and also on the power absorbed in a given diameter. It is necessary to specify the operating conditions in all comparisons. A typical set of efficiency curves is given in Fig. 4. The propulsive efficiency is plotted against the coefficient V/nD. It is to be noted that at a given V/nD the efficiency assumes many values, depending on the pitch. The maximum at a given V/nD is seen to be greatest for a propeller operating a little below its point of maximum efficiency.

Diagrams like Fig. 4 are not very convenient in actually selecting a propeller to fit a given condition. (The auxiliary curves of thrust and power coefficients required have been omitted here.) In practice the engine power and the revolution speed are specified, the forward speed is estimated, and the propeller pitch and diameter have to be determined. Another coefficient C_s from which the diameter is eliminated is most useful. The values of $C_S = \sqrt[5]{\frac{\rho V^5}{P n^2}}$ are plotted against propulsive efficiency and V/nD in Fig. 5 using the same data as for Fig. 4. The construction and use of this diagram is fully explained,[13] and it is sufficient to say here that the highest efficiency obtainable at a given operating condition, namely, a given C_s, lies on the envelope of the efficiency curves for the several pitch settings. When comparing various propeller-body arrangements it is sufficient to plot the envelope-efficiency curve against C_s. If the propeller diameter is also to be estimated, the auxiliary lines of V/nD against C_s shown at the bottom of Fig. 5 are, of course, required.

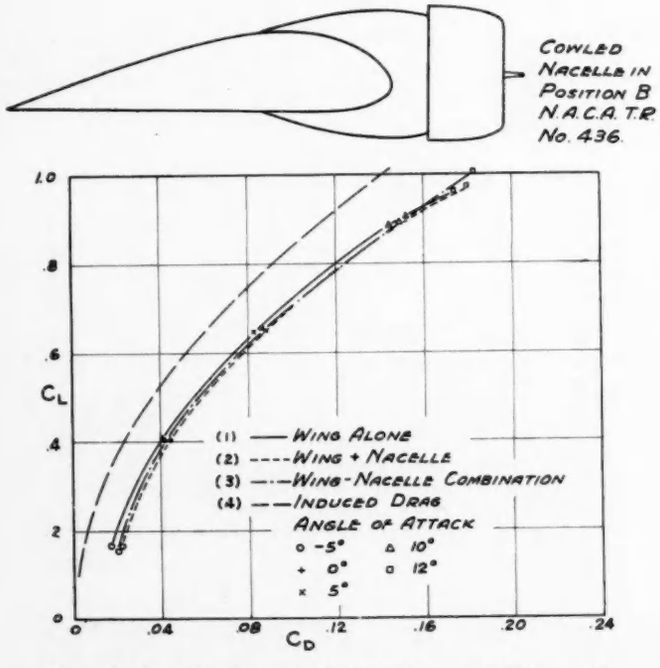

Fig. 8—Nacelle Drag and Interference; Wing with Aspect-Ratio 3; Cowled Nacelle Ahead of Wing

Solution of Example I

The solution of the example problem can now be continued. Repeating equation (1a)

$$\left(C_{D_P} + C_{D_N}\right) S \times \frac{\rho V^3}{2} + \frac{2 \times (\text{lift})^2}{\pi \rho V b^2} = P \times \eta \qquad (1a)$$

This equation is conveniently solved for V by trial and error. Usually a good guess can be made at the value of V at the beginning, and a second trial usually gives the solution almost exactly. For this case, a speed $V = 128.5$ m.p.h. $= 188.5$ ft. per sec. is assumed.

Now

$C_{D_P} = 0.0344$ Lift = weight = 12,125 lb.

$C_{D_N} = 0.0123$ Span = b = 75.5 ft.

$\rho = 0.002378$ slug per cu. ft. (standard sea-level density)

$P = 2 \times 555 = 1110$ hp. $n = \frac{1950}{60}$ r. p. s.

Then

$$0.0344 + 2 \times 0.0123 \times 882.6 \times \frac{0.002378}{2} \times (188.5)^3 + \frac{2 \times (12{,}125)^2}{\pi \times 0.002378 \times 188.5 \times (75.5)^2} = 1110 \times 550 \times \eta$$

$$\frac{414{,}000 + 36{,}600}{550} = 1110 \times \eta \qquad (1b)$$

C_s is now determined thus:[13]

$$C_S = \sqrt[5]{\frac{\rho V^5}{P n^2}} \text{ or } \frac{0.638 \times V}{P^{1/5} n^{2/5}} \text{ (for standard sea-level density)}$$

Then

$$C_S = \frac{0.638 \times 128.5}{3.54 \times 20.7} = 1.119$$

The propulsive efficiency has not been determined for the range of pitch settings exhibited in Figs. 4 and 5 for the nacelle position (Fig. 3) now being discussed. The range covered is, however, sufficient for the present problem, and the envelope efficiency is given as curve (1) of Fig. 6. At a value of $C_s = 1.119$ just determined

$$\eta = 0.737$$

Then (1b) becomes

$$820 = 1110 \times 0.737 = 818$$

This computation shows that the assumed speed is correct for all practical purposes (within about 0.1 per cent).

It is interesting to see how the different drag factors contribute to the performance (or lack of it). The values given in Table 1 are calculated from the usual formulas

$$\text{Drag} = C_D S q = C_D S \frac{\rho V^2}{2}$$

$$\text{Hp.} = \frac{\text{drag (lb.)} \times \text{velocity (m. p. h.)}}{375}$$

It will be noted that the nacelle drag alone accounts for 30.2 per cent of the drag and uses 22.3 per cent of the engine

Table 1—Data from Examples I, II, and III

No.		Drag	Nacelles Fig. 3 Example I		Nacelles Fig. 7 Example II		Nacelles Fig. 8 Example III	
			Lb.	Per Cent	Lb.	Per Cent	Lb.	Per Cent
(1)		Profile (C_{D_P} = 0.0344)	1280	53.6	1607	75.0	1870	84.7
(2)		Nacelle	722	30.2	236	11.1	275	12.5
(3)		Nacelle Interference	193	8.1	142	6.6	−73	−3.3
(4)	(2)+(3)	Effective Nacelle	915	38.3	378	17.7	202	9.2
(5)		Induced	195	8.1	155	7.3	134	6.1
(6)	(1)+(4)+(5)	Total	2390	100.0	2140	100.0	2206	100.0
		Power	B. Hp.	Per Cent	B. Hp.	Per Cent	B. Hp.	Per Cent
(7)		Profile Drag	439	39.5	617	55.6	773	69.6
(8)		Nacelle Drag	248	22.3	90.5	8.2	114	10.3
(9)		Interference Drag	66	5.9	54.5	4.9	−30	−2.7
(10)	(8)+(9)	Effective Nacelle Drag	314	28.2	145	13.1	84	7.6
(11)		Induced Drag	67	6.0	59.5	5.4	56	5.1
(12)	(7)+(10)+(11)	Thrust b. hp.	820	73.7	821.5	74.1	913	82.3
(13)	(1−η) b. hp.	Propeller Loss	292	26.3	288.5	25.9	197	17.7
(14)		Total	1112	100.0	1110	100.0	1110	100.0
(15)		C_L	0.327	....	0.259	....	0.224	....
(16)		C_{D_N}	2×0.0123	....	2×0.00405	....	2×0.00184	....
(17)		C_{D_T}	0.0642	....	0.0458	....	0.0407	....
(18)		L/D	5.1	....	5.66	....	5.5	....
(19)		C_S	1.119	....	1.255	....	1.350	....
(20)	(η) Propulsive Efficiency		....	73.7	....	74.1	....	82.3
(21)	Nacelle Drag Efficiency Factor		....	28.2	....	13.1	....	7.6
(22)	Net (Nacelle-Propeller) Efficiency		....	45.5	....	61.0	....	74.7
(23)	High Speed (m.p.h.)		128.5	....	144	....	155	....

power. The nacelle interference drag accounts for 8.1 per cent of the drag and uses 5.9 per cent of the power. The nacelle drag and interference (i.e., the effective nacelle drag) accounts for 38.3 per cent of the drag and uses 28.2 per cent of the engine power. The latter value is nothing but the nacelle-drag efficiency factor (N. D. F.) discussed in the N.A.C.A. references. The propulsive efficiency, (0.737) minus N. D. F. (0.282) gives a net efficiency (0.455). This net efficiency is the fraction of the motor power remaining after accounting for the nacelle drag and interference and the propeller losses. This is the fraction of the engine power that is available for overcoming the profile and induced drag of the other parts of the airplane, and for climbing or accelerating under conditions other than the high speed. As pointed out earlier this net efficiency is very useful in making comparisons.

Example II

As a first step in improving the performance of the example airplane, the engine will be assumed to be cowled, and the nacelle faired into the wing. The new nacelle arrangement is indicated in Fig. 7. The wing-alone, wing + nacelle, and wing - nacelle combination polar diagrams are given in the same form for this case as in the first example (Fig. 3). Some improvement is to be expected from the change in nacelle cowling, and the difference in drag is read at a lower value of the lift coefficient, where $C_L = 0.26$:

Model $C_{D_N} = C_{D_C} - C_{D_W} = 0.302 - 0.0236 = 0.0066$

and scaled to full size as in Example I.

$$C_{D_N} = 0.00405$$

The propulsive efficiency is determined from curve (2) of Fig. 6, which applies to the nacelle arrangement indicated

Fig. 9—Airplane with Modern Engine Nacelles

in Fig. 7. Values of the factors calculated in the same manner as has been indicated in Example I are given in Table 1.

The nacelle drag is greatly reduced by cowling as also is the interference drag so that the power used by the effective nacelle drag is reduced from 28.2 per cent of the engine power to 13.1 per cent. The propulsive efficiency is increased slightly (0.4 per cent). The net result is an increase in speed of 15.5 m.p.h., or 12 per cent.

Example III

As a third example a cowled nacelle is assumed to be located ahead of the wing as indicated in Fig. 8. The polar diagrams are here given as in the previous examples and, with $C_L = 0.220$

Model $C_{D_N} = C_{D_C} - C_{D_W} = 0.0230 - 0.0200 = 0.0030$

Full size $C_{D_N} = 0.00184$

The propulsive efficiency is determined from curve (3) of Fig. 6, which has been redrawn from Fig. 5. Again the calculations are carried through, and the resulting values form the fifth and sixth columns of Table 1.

The nacelle drag is greater than in Example II because of the higher speed. The interference drag is negative, however, and the fraction of the engine power used in nacelle drag is reduced to 7.6 per cent. The propulsive efficiency is 8.2 per cent higher than in Example II, and the speed has increased 11 m.p.h., or 7.6 per cent.

The improvement over Example I is a 78 per cent reduction in effective nacelle drag; a saving of 73 per cent in the engine power used by effective nacelle drag, and a 20 per cent increase in high speed (128.5 to 155 = 26.5 m.p.h. increase). The saving in effective nacelle drag is made up of a 62 per cent saving in actual drag due to cowling and a 138 per cent reduction in interference drag. The interference drag in Example I is 26.7 per cent of the nacelle drag, positive; and in Example III is 26.5 per cent, negative.

Table 2—Data from Examples IV and V

No.		Drag	Nacelles Fig. 3 Example IV		Nacelles Fig. 8 Example V	
			Lb.	Per Cent	Lb.	Per Cent
(1)		Profile (C_{D_P}=0.020)..	1181	43.5	1816	78.6
(2)		Nacelle.....	1060	39.0	429	18.6
(3)		Nacelle Interference...	261	9.6	−76	−3.3
(4)	(2)+(3)	Effective Nacelle.....	1321	48.0	353	15.3
(5)		Induced.....	217	8.0	141	6.1
(6)	(1)+(4)+(5)	Total.....	2719	100.0	2310	100.0
		Power	B. Hp.	Per Cent	B. Hp.	Per Cent
(7)		Profile Drag.....	491	34.6	936	65.9
(8)		Nacelle Drag.....	440	31.0	221	15.5
(9)		Interference Drag....	109	7.6	−39	2.7
(10)	(8)+(9)	Effective Nacelle Drag	549	38.6	182	12.8
(11)		Induced Drag	90	6.3	73	5.2
(12)	(7)+(10)+(11)	Thrust b. hp.....	1130	79.5	1191	83.9
(13)	(1−η) b. hp.	Propeller Loss.....	290	20.5	229	16.1
(14)		Total.....	1420	100.0	1420	100.0
(15)		C_L.....	0.296	....	0.193	....
(16)		C_{D_N}.....	2×0.0112	....	2×0.00194	....
(17)		C_{D_T}.....	0.0461	....	0.0254	....
(18)		L/D.....	6.43	....	7.60	....
(19)		C_S.....	1.65	....	2.045	....
(20)	(η)	Propulsive Efficiency.....	....	79.5	....	84.0
(21)		Nacelle Drag Efficiency Factor.....	....	38.6	....	12.8
(22)		Net (Nacelle-Propeller) Efficiency...	....	40.9	....	71.2
(23)		High Speed (m.p.h.)..	156	....	193.5	....

Modern Airplane, Examples IV to VII

Changes in the art have been so rapid that Examples IV to VII are, in the light of present types, of "ancient vintage." Consideration of corresponding effects on a modern airplane is desirable. Accordingly, an airplane of the type illustrated in Fig. 9 will be examined. This airplane is assumed to have the following characteristics:

Total weight	17,500 lb.
Wing area	948.6 sq. ft.
Wing span	85 ft.
Engine power—2 engines, each developing 710 hp. at 1,950 r.p.m., 53¾ in. in diameter, geared 11:16	
Profile-drag coefficient (without nacelles)	0.020
Wing loading	18.45 lb. per sq. ft.
Power loading	12.33 lb. per hp.

Although the photograph of the airplane makes it appear difficult, it will be assumed for illustration that uncowled nacelles similar to Fig. 3 are installed. Following the methods already outlined, the data of Table 2 are compiled in the columns marked Example IV. Alternately cowled nacelles as in Fig. 8 are assumed and the data listed under Example IV are obtained.

These two examples are calculated for standard sea-level density (ρ = 0.002378 slug per cu. ft.). At the present time air transports are designed for, and operate normally at, a moderate altitude. The calculations have, therefore, been repeated for a standard altitude of 8,000 ft. (ρ = 0.001869 slug per cu. ft.), assuming that the engines are supercharged to give the same power. This assumption is slightly unfavorable to a proper comparison because an engine designed for altitude usually develops somewhat less power at ground level when operated as recommended by the manufacturer. The advantage of comparing at the same power, however, outweighs any disadvantages in this case. The data for the altitude performance are given in Table 3 under Examples VI and VII.

The airplane of Examples IV to VII is much "cleaner" than that of Examples I to III, as manifested by the 42 per cent lower profile-drag coefficient. The engines have the same linear dimensions in all cases, so that with the reduced profile drag coefficient the effective nacelle drag becomes a larger proportion of the total drag, about 10 per cent more in Examples IV and VI than in Example I, and about 6 per cent more in Examples V and VII than in Example III. The increase in speed due to nacelle change is 37.5 m.p.h. at sea level and 40.5 m.p.h. at 8,000 ft. for the modern plane, compared to 26.5 m.p.h. for the older airplane at sea level. The improvement in the older airplane performance is considerably aided by the 8.6 per cent increase in propulsive efficiency. On the other hand, the increase in propulsive efficiency on the modern airplane due to nacelle change is only 4.0 per cent. An increase of 1 per cent, however, means more miles per hour on the higher-speed plane. The change in engine nacelles indicates that about 73 per cent of the power formerly used in nacelle drag can be saved on an older type airplane of the characteristics given (i. e. a saving of

230 hp.). A corresponding saving of about 67 per cent (i. e., about 365 hp.) is indicated for the modern airplane. As given in the examples, these power savings were utilized to obtain higher speeds but, if higher speeds were not required, either the power of the airplane could be reduced or greater loads carried. Some idea of the relative overall efficiencies of the old and new airplanes is given by the values in the following table:

Example	$\frac{\text{Weight (lb.)} \times \text{speed (m.p.h.)}}{\text{Engine power (b.hp.)}}$	Relative values	
I	1,400		1.00
II	1,573		1.12
III	1,693		1.21
IV	1,923	1.00	1.37
V	2,385	1.24	1.70
VI	2,070	1.00	1.48
VII	2,570	1.24	1.83

These values indicate about an 83 per cent improvement in airplane design performance, of which about 24 per cent is due to improved cowling and better location of engine nacelles. The old airplane carried a payload of about 21 per cent, and the modern airplane carries about 18.5 per cent, but the latter has about twice the range, even at full power. On an economic basis the improvement is, therefore, greater than tabulated above.

Discussion of Examples

The examples just considered illustrate the improvement in high speed that can be obtained by proper cowling and location of the engine nacelles. The figures given must not be taken as all-inclusive, nor viewed with a security that may be indicated by the decimal places in some of the figures given in the tables. On a comparative basis they are considered correct and, quantitatively, they are close to performances that have been actually obtained.

Each case must be decided on its merits and the particular values obtained will depend on the airplane, the size and power of the engine, and the air speed. As mentioned earlier, quotation of values without qualification is misleading. Some have contended that practically all the improvement in airplane performance in recent years has been due to the higher wing loadings used, and not so much to refinement. The examples do not support this view. The contribution of reduced drag is very real.

Nacelle Drag.—The nacelle drag evidently depends on the engine size, and this effect will now be considered in greater detail. Other things being equal, an engine of smaller cross-sectional area will have a lower drag. Improved performance will result if engines of smaller dimensions are employed. Indeed, this consideration is undoubtedly one of the principal reasons for the development of two-row radial engines. In order to show the effect of engine size with any generality, a large number of additional examples would be required were it not for the fact that the net efficiency affords a convenient method of comparison.

The effective drag of engine nacelles of different sizes may be calculated on the assumption that the effective drag is proportional to the cross-sectional area. This assumption is admittedly an approximation because there is no guarantee that either the drag or interference vary in this way. There is some evidence, however, that this assumption is not far wrong and it offers the only method available at present. The effective drag of several engines has been calculated using coefficients for a cowled nacelle located in the position ahead of the wing (Fig. 8). These values have been reduced to nacelle-drag factors by multiplying the drag by the velocity and dividing by the engine power, and are plotted against velocity in the lower curves of Fig. 10. The characteristics of the engines are also listed. A representative constant coefficient (0.0026) has been used[1] (based on model wing area at $C_L = 0.25$) and the variation with lift coefficient (the lift coefficient will depend on the particular airplane) is ignored. This simplification does not affect the comparison. It is to be noted that these drag values show a variation as the cube of the speed and the spacing of the curves depends on the relative cross-sectional area of the different engines. Furthermore, each curve does not apply to one airplane but rather each point on a curve is the high speed of some airplane—a different airplane for each point.

Using the data of Fig. 6 giving the maximum propulsive efficiency obtainable with the nacelle-propeller arrangement chosen, a series of curves of propulsive efficiency against speed are plotted for the different engines. These are the upper curves of Fig. 10. The net efficiency (η — N.D.F.) is obtained directly and the values are given in Fig. 11.

Examination of Fig. 11 brings to light several interesting facts. First, the maximum values of net efficiency for the

$$[1]\text{Nacelle drag factor} = \frac{0.0026 \times 75 \times \frac{(\text{Eng. dia.})^2}{20^2} \times \frac{\rho V^2}{2} \times V}{\text{Engine power}} = \frac{0.0004875\ (\text{Eng. dia.})^2 \times q \times V}{\text{Engine power}}$$

Table 3—Data from Examples VI and VII

No.		Drag	Nacelles Fig. 3 Example VI Lb.	Nacelles Fig. 3 Example VI Per Cent	Nacelles Fig. 8 Example VII Lb.	Nacelles Fig. 8 Example VII Per Cent
(1)		Profile (C_{D_P}=0.020)..	1076	42.7	1656	77.6
(2)		Nacelle.....	967	38.4	391	18.3
(3)		Nacelle Interference...	238	9.5	−69	3.2
(4)	(2)+(3)	Effective Nacelle.....	1205	47.9	322	15.1
(5)		Induced.....	238	9.4	155	7.3
(6)		Total.....	2519	100.0	2133	100.0
		Power	B. Hp.	Per Cent	B. Hp.	Per Cent
(7)		Profile Drag.....	482	34.0	920	64.8
(8)		Nacelle Drag.....	433	30.5	218	15.3
(9)		Interference Drag....	107	7.5	−38	2.7
(10)	(8)+(9)	Effective Nacelle Drag	540	38.0	179	12.6
(11)		Induced.....	107	7.5	86	6.1
(12)	(7)+(10)+(11)	Thrust b. hp.	1129	79.5	1185	83.5
(13)	(1−η) b. hp.	Propeller Loss.....	291	20.5	235	16.5
(14)		Total.....	1420	100.0	1420	100.0
(15)		C_L.....	0.326	....	0.222	....
(16)		C_{D_N}.....	2×0.0112	....	2×0.00194	....
(17)		C_{D_T}.....	0.0468	....	0.0257	....
(18)		L/D.....	6.97	....	8.64	....
(19)		C_S.....	1.70	....	2.10	....
(20)	(η)	Propulsive Efficiency.....	....	79.5	....	83.5
(21)		Nacelle Drag Efficiency Factor.....	....	38.0	....	12.6
(22)		Net (Nacelle-Propeller) Efficiency...	....	41.5	....	70.9
(23)		High Speed (m.p.h.)..	168	....	208.5	....

different engines do not differ greatly but do occur at different speeds. A given engine and propeller is only working at its best at one speed. Second, the smaller engines are of limited application. The net efficiency tends toward zero at much lower speeds than for the larger engines. For example, no matter how many of engines (1) Fig. 10 were installed an airplane could never attain a speed of 240 m.p.h. Third, the importance of power in a given size is very evident. The 90-hp. engine is small but its power is also small for its size, and its drag results in a considerably lower net efficiency than that of the 225-hp. engine. Also, the two-row engine (5) is superior to the single-row engine (4) at speeds above 160 m.p.h., although a small part of this difference is due to slightly higher power of the two-row engine in these particular examples. Other engine-nacelle-propeller characteristics can be readily plotted in this way, and a good approximation of their relative merits obtained. The nacelle-drag factors (N.D.F.) may be determined with little calculation from the chart (Fig. 12), and the propulsive efficiencies, from Fig. 5. The curves of Fig. 12 are calculated for the same drag coefficient as used in Figs. 9 and 10.

It is evident that a particular engine can be adapted to some extent to a particular speed by a proper choice of the gear ratio to bring the propulsive efficiency nearer to a maximum at that speed. In general, this requires an impractically large number of gear ratios so that, practically, the maximum possible net efficiency can be obtained only at a limited number of speeds. It is to be expected, however, that the current gear ratios will change as may be required at higher speeds and altitudes.

Propulsive Efficiency.—It will have been noted, however, in Fig. 5 that the maximum propulsive efficiency falls off at pitch settings above about 27 deg. The particular propeller used throughout these investigations (No. 4412) has a pitch that is uniform with radius except near the hub when the propeller is set at 12 deg. When the propeller is set at higher angles, there is a wash-out of pitch toward the hub increasing with the setting. There is some evidence that the falling off in efficiency at the high-pitch settings can be largely avoided if the blade angles are laid out for a uniform pitch at a somewhat higher setting, say about 25 to 30 deg. The propellers now in common use have about the same pitch distribution as the No. 4412 described and might be improved by changes in pitch distribution. Although the loss is just beginning to be apparent at the pitch settings now used (about 34 deg. maximum), it will increase as higher speeds are attained. Experiments are soon to be made with full-scale propellers to provide more reliable data on the best pitch distribution as well as on the efficiency itself at higher settings. No full-scale data at pitches above 28 deg. are available at present.

Other Radial-Engine Nacelle Arrangements

The foregoing has been devoted to the discussion of radial engines for reasons stated at the outset. The less efficient arrangements have, however, been passed over. Before beginning a discussion of other engine types, it may be of interest to review some of the high lights of these other radial-engine installations. As far as the cowled nacelle is concerned, it has been shown to be practically as efficient on a biplane as on the monoplane.[2] The nacelle ahead of the upper wing seems to be slightly better than when ahead of the lower wing, although probably not enough to overcome its practical disadvantages. The radial engine does not show to advantage when operating a pusher propeller[12] because no cowling has been found that reduces its drag to anywhere near that of the N.A.C.A. cowled nacelle.[5] This difficulty is inherent in the close proximity of the propeller and engine cylinders which prevents the gradual pointing of the body at the rear that is essential for low drag. The tandem-engine-propeller arrangements suffer from disadvantages that are almost entirely due to the same difficulties of cowling the rear engine.[10] The pusher propeller has somewhat higher efficiencies, but not enough to overcome the higher drag, and it is likely to offer practical difficulties, owing to the beating of the blades as they pass the trailing edge of the wing. Observations in model tests indicate that the pusher propeller is considerably more noisy than the tractor. Altogether there is no question about the superiority of the cowled nacelle located ahead of the leading edge of the wing.

The In-Line Engine

The only other air-cooled engine that has attained any prominence is the in-line engine, which has until quite recently appeared mostly as a single-bank engine of low power. An air-cooled engine with four banks of cylinders arranged in the form of an H was announced a short time ago, and rumors are current of the development of single-bank engines of 500 hp. These developments in the direction of high power are a recognition of the fact that only the development of high power in reduced space, or rather in reduced exposed space, can improve on existing equipment. It is too early to make many predictions, but the few comparable experiments made indicate that the single-bank in-line engine offers a higher resistance than the cowled radial, which is partly compensated for by a higher propulsive efficiency. The fine performances put up by certain airplanes equipped with in-

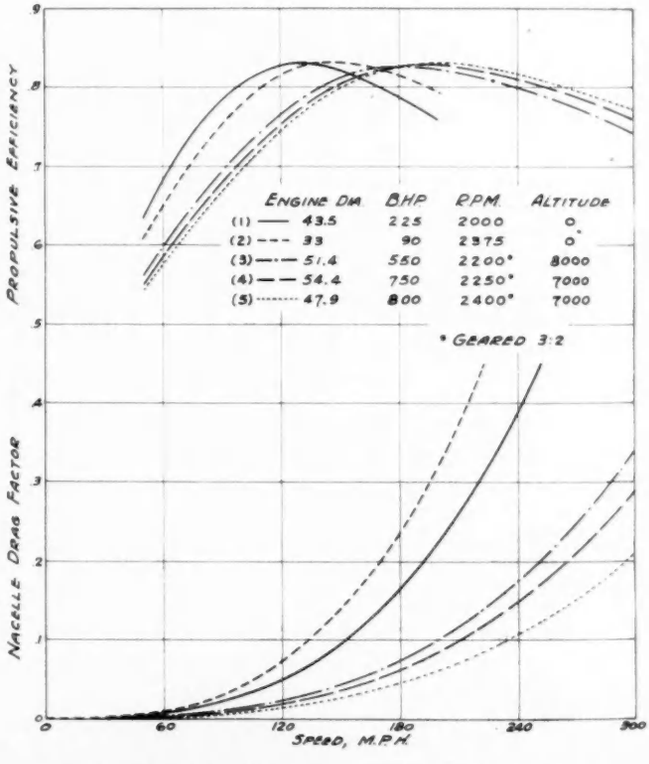

Fig. 10—Propulsive Efficiencies and Nacelle Drag Factors for Radial Engines

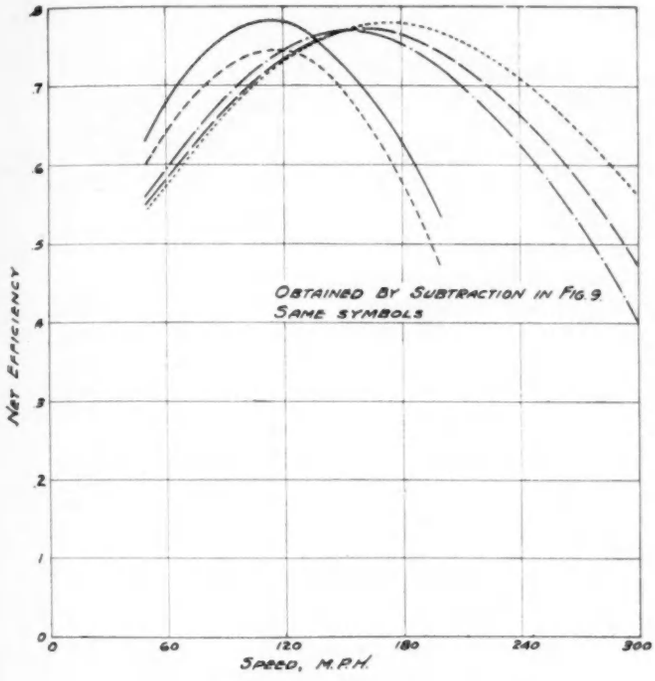

Fig. 11—Net Efficiencies for Radial Engines

line engines have been used to full advantage in advertising, but no tests with radial engines under comparable conditions have been made. No real evidence has been submitted that there is any great superiority of one type over the other.

If experience with the radial engine is worth anything, considerable design development to secure adequate cooling is to be anticipated when the horsepowers of in-line engines are increased. This work will require attention to the method of bringing in the air, of leading it around the cylinders, and of letting it out. It appears that difficulties are already at hand, and the N.A.C.A. proposes to study the cooling of the in-line engine in considerable detail before proceeding further with the question of its nacelle efficiency. The good looks of the present in-line engine housing may be somewhat altered in the process.

The Liquid-Cooled Engine

The liquid-cooled engine has survived many vicissitudes in the history of aeronautics and, though it has seemed to be in a state of suspended animation in many periods, it is still very much alive. Those who have crossed it off the list at one time have often been found among its advocates at a later date. In the nature of things, no one engine type can do everything. The air-cooled engine has done much but has not approached the power developed by the liquid-cooled engine, even granting that the highest powers in the latter have resulted from extreme boosts.

The relative merits of different engine types cannot be discussed here except as related to their aerodynamic efficiency. The liquid-cooled engine requires certain auxiliary apparatus called radiators, which both practically and aerodynamically are responsible for many of its failings. The engine itself can be housed in a smooth body of low drag, but then the radiators must be hung on 'round about (steam, water, or Prestone) or cleverly concealed in the walls of the engine housing (oil). The suspended radiator is responsible for a great deal of drag, and the concealed radiators are not very efficient. In a recent full-scale test the radiators (Prestone and oil) were found to be responsible for 11.7 per cent of the total airplane drag (airplane speed 188 m.p.h.). A test with a model wing and nacelle showed a drag for the suspended radiator alone equal to the drag of the nacelle, the whole (nacelle plus Prestone and oil radiators) accounting for about 15 per cent of the total airplane drag for a 200-m.p.h. airplane of the type considered in the examples previously given.

These values are comparable with results for radial air-cooled engines and are in agreement with flight tests, showing that there is no appreciable superiority of one type over another. If anything, there is a slight advantage in favor of the liquid-cooled engine.

Possibility of Improvement.—The present liquid-cooled engine installations are improvements over earlier types in that steam and Prestone have been substituted for water as the cooling liquid with the result that smaller radiators are permitted. Retractable radiators are also employed to some extent. A housing or cowling has also come to be applied to most radiators. It was discovered many years ago that a slight projection of the shell ahead of the core increased the cooling performance of the radiator. Probably, little attention was then paid to the radiator drag.

In the author's opinion the whole radiator question requires a complete overhaul. There is little doubt that the drag of radiators can be greatly reduced by proper cowling. The problem is analogous to that of cooling the air-cooled engine but is a little more complicated. A given engine has fixed dimensions, and the problem is to cool it with the least drag; whereas, the radiator dimensions are subject to variations to secure the required heat dissipation with the least drag. Tests without number of the heat dissipation of radiator cores have been made. Most of these are quite useless because the type of flow into an isolated core is radically different from that through a core mounted alongside of a fuselage or nacelle with cowling around it.

Recent experiments on the cooling of air-cooled engines have shown that the mechanism of cooling is more complicated than was previously imagined; yet the efficiency with which cooling is accomplished is better than anticipated, in fact, better than some theoretical maximum estimates. The radiator problem should be approached with an open mind and without any preconceived notions. It will not be surprising if the radiator drag can be reduced to less than half of its present value. The liquid-cooled engine will then be-

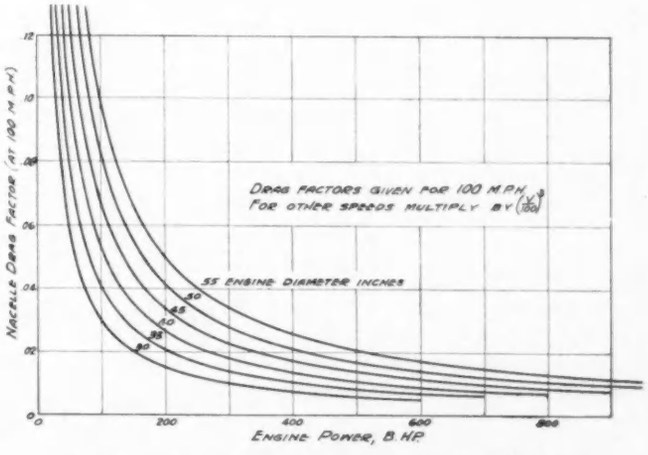

Fig. 12—Chart for Use in Calculating Nacelle Drag

come a most serious competitor of the air-cooled engine in the high powers that are just ahead.

Other Possibilities

The engine nacelle is nothing more than an appendage necessitated by the engines that drive the propellers. It is not surprising, then, that consideration has been given to designs in which the engines are placed within the wing and the propeller driven by a shaft, which can be housed in a small covering. These proposals are in accordance with others reducing the airplane to a mere wing which, after all, does the work of carrying the load. The elimination of all unnecessary appendages is in the interest of higher efficiency and is to be recommended provided that undesirable features are not introduced in the process.

Some people have considered that other types of engines are better adapted to installations in a wing than present types. Flat engine designs and many others have been proposed. Irrespective of the type of engine, some means of cooling must be provided and, in order not to increase the drag and thus decrease the advantage of the extension shaft arrangement, great care must be used in the cooling-system design. One possibility is the surface or "skin" radiator. This design has been tried and discarded but probably will be revived and developed further if sufficient impetus is given.

Development of the shaft will also be required to provide proper support and stiffness to insure that undesired vibrations do not occur as a result of resonances with the propeller vibration frequencies and that the important critical speeds are outside the operating range. The successful operation of some designs of this type has already shown that these problems are not insurmountable.

Extension Shaft Experiments.—A few experiments, comparable with those previously described, have been made to determine the net efficiencies when extension shafts are used. The arrangements tested ((1) to (4)) are shown at the top of Fig. 13. Some question may be raised regarding these particular shapes of housing. The present discussion is intended only as an outline of the possibilities. Other experiments with additional variations in shape are shortly to be made. The nacelle drag factors and net efficiencies given are calculated from the measured effective-nacelle drag for an engine developing 700 hp. at 2,500 r.p.m. at sea level and geared 3:2.

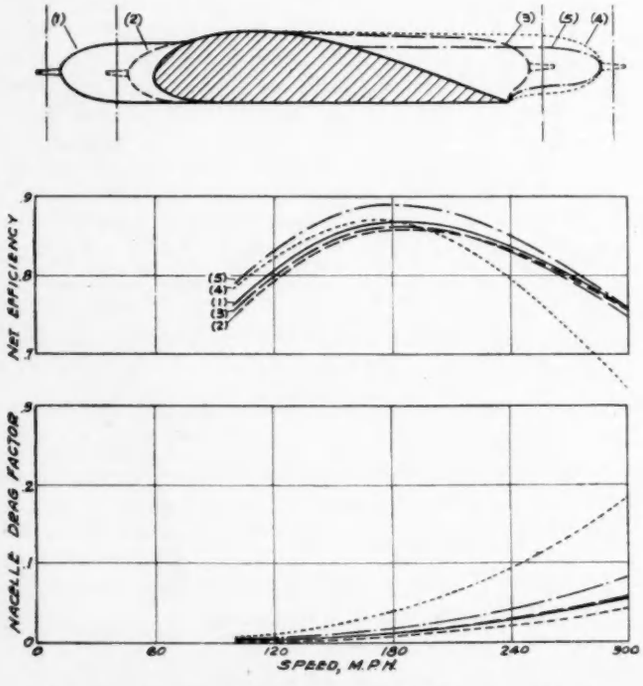

Fig. 13—Nacelle Drag Factors and Net Efficiencies for Engines with Extension Shafts

The maximum diameter of the shaft housing is 27 in. full size. This shaft housing is larger than might be required and the lines (5) have been drawn in to represent the minimum practicable size on the assumption that there is a ball-bearing immediately behind the propeller. This housing has a maximum diameter of 18 in. The nacelle drag factor is reduced in proportion to the ratio of the diameters[2] from curve (4).

The net efficiencies are given by the upper curves of Fig. 13. It will be noted that the forward tractor position (1) gives the highest net efficiency at speeds above 185 m.p.h. of the arrangements (1) to (4) actually tested. The drag is nearly as low as for (2) and the propulsive efficiency is higher. The higher propulsive efficiency of the pusher arrangements (3) and (4) is insufficient to overbalance their higher drag. The importance of the drag is clearly shown by the drop of the curve (4) above 170 m.p.h. and by curve (5) for the reduced-size shaft housing, the net efficiency for this case being above that for the other arrangements. It should be remarked that, if the same shaft size reduction were applied to the tractor arrangement (1), it would be the best at speeds above 250 m.p.h.

These arrangements may be compared with those for the radial engine (Fig. 11). At the higher speeds (above 200 m.p.h.) the 800-hp. radial engine 47.9 in. in diameter shows the highest net efficiencies.

At 200 m.p.h. the net efficiency is about 0.765. For the shaft arrangement (5) the net efficiency is 0.885 at the same speed for sea-level conditions. Converting this to 7,000 ft. to compare with the radial engine gives an efficiency of 0.89. This is 12.5 per cent higher than for the radial-engine nacelle. At higher speeds the difference is even greater; 15 per cent at 250 m.p.h. and 19 per cent at 300 m.p.h., although the actual efficiencies are successively lower.

Referring again to Examples V and VII, the net efficiency is about 0.71. This will increase to 0.885 with the shaft arrangement, a gain of 17.5 per cent equivalent to at least an 11 m.p.h. increase in speed or 248 hp. available for other purposes. Of the 17.5 per cent gain about 10.5 per cent is due to reduced nacelle drag and the remaining 7 per cent, to increased propulsive efficiency so that the shaft scheme has advantages other than drag reduction. It may be mentioned that about 5.5 per cent could be saved by the use of smaller radial engines instead of the large ones assumed in the examples so that the shaft-extension arrangement is about 12 per cent better than present best radial-engine nacelles.

It must be borne in mind that the foregoing results are predicated on the use of surface radiators causing no drag. It is clear from what has already been said about present-day radiators that their drag will more than make up for the advantages. For the engine installation in the wing to prove superior to existing arrangements, development along the lines already indicated is required. The advantages are so obvious that it is to be hoped that experiments will proceed without delay. Practical success will be awaited with interest by all those concerned with the efficiency of airplanes.